THE
INSTITUTION OF
AUTOMOBILE ENGINEERS

REPORT OF THE EMPIRE MOTOR FUELS COMMITTEE

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THE

Institution of Automobile Engineers.

REPORT

OF THE

EMPIRE MOTOR FUELS COMMITTEE

EMBODYING OTHER ALLIED RESEARCHES.

VOLUME XVIII.

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INTRODUCTION.

It is a matter of regret to the Committee which was entrusted with the task of the publication of this Report that it has found itself unable to devise a fully adequate title for this volume, including, as it does, so many other valuable data beyond those contained in the report itself. Before the question of publication had even begun to be seriously considered, it became evident that the value of the Report to some extent depended upon, and would at all events be greatly enhanced by, the inclusion of the results of other investigations, both theoretical and practical, which were available in various more or less inaccessible forms. The volume therefore opens with an account of the joint theoretical work of Mr. H. T. Tizard, M.A., and Mr. D. R. Pye, M.A., which formed the basis of the investigations subsequently carried out by Mr. H. R. Ricardo for the Asiatic Petroleum Co., Ltd., the results of which form the second Section of the book. The third Section is entirely occupied by the Report of the Empire Motor Fuels Committee, while the fourth Section is devoted to a series of complementary notes, designed to bring existing knowledge up to date, as follows :—

1. A note by Dr. A. E. Dunstan, D.Sc., F.I.C., M.I. Pet. Tech., on the investigations carried out in America in regard to the subject of detonation, which was brought into such prominence in the course of the experiments on alcohol.

2. A note by Dr. W. R. Ormandy, D.Sc., F.I.C., M.I. Pet. Tech., on the Physical and Chemical Properties of Alcohol in Admixture with other Fuels.

3. A note by Mr. H. R. Ricardo, B.A., M.I.A.E., on further researches carried out by him on several problems which were brought to light, but not solved, during his investigations published in Sections II. and III.

The volume may therefore be said to contain within its covers a complete review of the subject of fuels for use in high-speed internal combustion engines, so far as it is known to-day (July, 1924), and the data now placed on record are bound to be of great technical and practical value, alike to automobile engineers, to the producers of motor fuels, and to those interested in alternative or supplementary fuels.

The cost of the investigations carried out by Mr. Ricardo, as set out in Section II., was entirely borne by the Asiatic Petroleum Co., Ltd.; the cost of the investigations set out in Section III. was met by grants, including £312 10s. from the Department of Scientific and Industrial Research, £750 from the Royal Automobile Club and £750 from the Commercial Motor Users Association (Incorporated).

The cost of publication of the present volume has been met jointly out of the unexpended balance of the sum subscribed for the investigation contributed by the Empire Motor Fuels Committee, namely £200, a grant of £100 from the Institution of Petroleum Technologists, a grant of £100 from the Institution of Automobile Engineers, and the balance by a contribution from the Distillers Co., Ltd., while the Institution of Automobile Engineers has contributed a further share in bearing the burden of the editorial work.

The Empire Motor Fuels Committee, to which the credit for the inception of the idea of the publication of the Report is due, was set up by the Imperial Motor Transport Council, of which H.R.H. Prince Arthur of Connaught is President, and the Hon. Sir Arthur Stanley, G.B.E., is Chairman, in May, 1920, with the following terms of reference :—

(*a*) To take immediate steps to encourage and develop production and utilisation of additional motor fuel supplies or raw materials therefor in all parts of the Empire, and more particularly to ensure increasing shipments of motor fuels to Great Britain, and so far as may be necessary to the Empire Overseas.

(*b*) To co-operate with any Government or incorporated or unincorporated body of persons in order to further the object set forth in (*a*) hereof.

(*c*) To offer for the purposes in view bonuses or other rewards, and to incur such expenditure as may be desirable, all such expenditure being subject to the approval of the Council.

The composition of the Committee is as under :—

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Dr. W. R. Ormandy, D.Sc., F.I.C., M.I.A.E.
W. H. Ross, O.B.E.

Alcohol Excise Restrictions Sub-Committee.

Sir Charles Bedford, LL.D., D.Sc. (Chairman).
Dr. W. R. Ormandy, D.Sc., F.I.C., M.I.A.E.
W. H. Ross, O.B.E.

The first Sub-Committee was entrusted with the duties of preparing a synopsis regarding the development and use of power-alcohol engines, embodying information as to the lines of procedure with regard to experiments and including recommended standards for denaturants and mixtures.

The second undertook the study of new methods of denaturation for adoption throughout the British Empire, and considering the adoption of alternative methods of denaturation common to the whole Empire and accepted as such by the different Customs Authorities throughout the Empire.

The third was concerned with the modifications of existing Excise restrictions on the manufacture of industrial alcohol and on its carriage and storage in bulk.

The reports of the second and third Sub-Committees have been separately circulated.

The researches carried out at the request of the Departmental Committee appointed in 1918 by Mr. Walter Long under the direction of Prof. H. B. Dixon upon the ignition temperatures of alcohol and other vapours, and detonation of alcohol, etc., under various conditions, are referred to at length in the Proceedings of the Imperial Motor Transport Conference of 1920, pp. 77 *et seq.*

(*Signed*) W. R. ORMANDY,
Chairman of Publication Committee.

The Empire Motor Fuels Committee as a body desires to place on record :—

(1) Its high appreciation of the successful labours carried through by the Sub-Committees mentioned above.

(2) Its obligation to the Hon. Secretary of the Committee, Mr. Bristow, for his services during the past four years.

(3) Its warm appreciation of the generous action of the Asiatic Petroleum Co. in allowing the publication of such valuable scientific results obtained at great expense to themselves.

(4) Its thanks to Messrs. Tizard and Pye for kindly allowing their work to be embodied in this volume, to the proprietors of *The Automobile Engineer* for permitting the whole of the work published in that journal by Messrs. Tizard and Pye and Mr. Ricardo to be reproduced in this volume, and to Dr. A. E. Dunstan, Dr. W. R. Ormandy and Mr. H. R. Ricardo for compiling the final notes which form such a valuable Appendix to the volume.

(5) Its indebtedness to the Department of Scientific and Industrial Research, the Royal Automobile Club, the Institution of Petroleum Technologists, the Commercial Motor Users Association, The Distillers Co., Ltd., and the Institution of Automobile Engineers for the financial support already recorded.

(6) Its cordial recognition of the valuable help in editing the various articles and checking the figures throughout rendered by the Institution of Automobile Engineers, and particularly by Dr. Ormandy and its Secretary, Mr. Joy.

(*Signed*) E. S. SHRAPNELL-SMITH,
Chairman, Empire Motor Fuels Committee.

London, 24*th June*, 1924.

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SECTION I.

THE CHARACTER OF VARIOUS FUELS FOR INTERNAL COMBUSTION ENGINES.

THE CHARACTER OF VARIOUS FUELS FOR INTERNAL COMBUSTION ENGINES.

The Influences of Specific Heat and Dissociation of the Working Fluid.

By H. T. Tizard, M.A., and D. R. Pye, M.A., A.I.A.E.

Part I.

The following theoretical work was undertaken as a preliminary to the complete experimental investigation into the effect of chemical composition of the fuel upon engine performance by Messrs. Ricardo, in which the authors have had the privilege of taking some part.

The whole of the work was undertaken for the Asiatic Petroleum Co., who agreed that the investigation should not be confined to matters of possible commercial value, and throughout they have, very generously, never raised the smallest objection to the publication of any results of general scientific interest which have arisen in the course of the work.

I.

In seeking for a "standard engine of comparison" with which the calculated performance of actual internal combustion engines might be compared, the earliest proposal was that the standard engine should be supposed to follow the "constant-volume" cycle, with "ideal air" as the working substance.

"Ideal air" is a perfect gas, having constant specific heats at all temperatures, for which the value of γ, the ratio of the specific heats at constant pressure and constant volume is equal to 1.4.

In an engine working on the above cycle the working substance is supposed to be :

(1) Compressed adiabatically in the ratio $\frac{V_A}{V_B} = \gamma$.

(2) Instantaneously heated at constant volume $V_B = V_C$ to its maximum temperature t_C° C.

(3) Expanded adiabatically to its original volume $V_D = V_A$.

(4) Cooled instantaneously at constant volume to its original temperature t_A° C.

Such a cycle has the advantage that the thermal efficiency of an engine working upon it is very simply expressed as

$$1 - \left(\frac{1}{r}\right)^{\gamma - 1}$$

The thermal efficiency therefore depends only on the ratio of compression, and will steadily increase as the compression is raised.

There are different senses in which the "standard engine of comparison" may be looked upon as the ideal with which real and imperfect engines may be compared. There is an essential difference, for instance, between an engine of comparison which postulates a working substance such as "ideal air" and one which takes account of the properties of the real working substance and only defines and simplifies the working conditions by assuming, for instance, that the heating of the working substance is instantaneous, and that there shall be no loss or gain of heat by conduction through the cylinder walls. The first, from the nature of things, must ever remain far removed from practical possibilities, while the second, by taking into account the fundamental physical conditions and properties of the substances involved in the working of an actual engine, becomes an approachable, though not, so far as can be seen, an attainable, ideal.

It has been recognised for some time that the "ideal air" cycle is sometimes too far removed from practical possibilities to form a satisfactory basis of comparison. An ideal cycle should afford a guide to engine designers in the sense that it should show what is to be expected as a result of a modification in design, such as raising the compression-ratio, any fundamental physico-chemical conditions being taken into account, but the processes in the engine being legitimately simplified by the neglect of practical imperfections, so as to allow results to be calculated with reasonable ease and to make the conditions of calculation independent of any particular engine.

In accordance with the foregoing principles, therefore, we should, in defining the cycle of operations for the standard engine of comparison, take into account the properties of the real working substance (a mixture, as a rule, of N_2, CO_2, H_2O, with smaller quantities of CO, H_2, O_2, and other compounds), and any fundamental conditions which must obtain in the cylinder during and after combustion.

II.

The power and efficiency of an internal combustion engine may vary with the type of fuel used. The most common liquid fuel is a mixture of hydrocarbons, which may vary in chemical composition from those containing a high proportion of carbon—*e.g.*, the aromatics C_nH_{2n-6}—to those containing a relatively much higher

proportion of hydrogen—*e.g.*, the paraffins C_nH_{2n+2}. There is no doubt that chemical composition has a profound effect on behaviour in an engine from some points of view, such as tendency to detonation, but, apart from this, the question arises whether by a change of fuel the maximum power of an engine, working under otherwise constant conditions, can be increased. It is clear that the factors controlling maximum power, assuming equal volumetric efficiency, are :—

(*a*) Heat of combustion per *unit volume* of combustible gas.
(*b*) Change in the number of molecules on combustion.
(*c*) Specific heat of the resulting gases.

Item (*a*) can be calculated for pure hydrocarbons from the chemical equation of the reaction if we know the molecular heat of combustion, or the heat of combustion per gram. Thus the heat of combustion of benzene, including the heat of condensation of the water formed, is known from the very accurate researches of T. W. Richards* and his collaborators to be 780 kg. calories per gram molecule. The *lower* molecular heat of combustion is therefore 780,000 $- 54 \times 540 = 751$ kg. calories. The combustion equation is

$$2\ C_6H_6 + 15\ O_2\ (+ 56.4\ N_2) = 12\ CO_2 + 6\ H_2O\ (+ 56.4\ N_2).$$

Total molecules before reaction = 73.4.
Total molecules after reaction = 74.4.
Ratio = 1.013.

Heat of combustion per *gr. molecule* of combustible mixture

$$= \frac{2 \times 751}{73.4}$$

$$= 20.46 \text{ kg. calories.}$$

Richards's experiments, which were made with benzene, toluene, the three isomeric xylenes, the five isomeric octanes, and cyclohexane, are of a very high order of accuracy. The results of other workers on the heats of combustion of pure hydrocarbons are not so accurate, but sufficient data exist to show that in any series of hydrocarbons an addition of a CH_2 group to the molecule, *i.e.*, an addition of 14 to the molecular weight, increases the (lower) molecular heat of combustion by about 145 kg. calories. The figures given in Table I. are considered the best available—those not due to Richards are taken, for the most part, from Landolt and Bornstein's Tables (1912 edition).

It is interesting to note that although the heat of combustion per gram (the so-called calorific value) is about 10 per cent lower for aromatic than for paraffin hydrocarbons, and for ethyl alcohol is only two-thirds of the value for benzene, yet the heat of combustion

* *J. Amer. Chem. Soc.*, 1910, Vol. 32, p. 268; 1915, Vol. 37, p. 993; 1917, Vol. 39, p. 341.

TABLE I.

Hydrocarbon.	Formula.	Boiling point.	Heat of combustion in kg. Cals. A. Per gramme of fuel.	B. Per gm. molecule of combustible mixture.	C. Volume ratio.	B×C.
Paraffins.						
Hexane	C_6H_{14}	69·0	10.75	20.03	1.053	21.1
Heptane	C_7H_{16}	98.4	10.7	20.1	1.056	21.2
Octane	C_8H_{18}	124.5	10.65	20.10	1.058	21.2
Aromatics.						
Benzene	C_6H_6	80.4	9.64	20.46	1.013	20.7
Toluene	C_7H_8	110.3	9.76	20.46	1.023	20.9
Xylene	C_8H_{10}	140 (abt.)	9.85	20.44	1.029	21.1
Naphthenes.						
Cyclohexane ..	C_6H_{12}	80.8	10.5	20.09	1.045	21.0
Ethyl alcohol ..	C_2H_5OH	78.3	6.45	19.41	1.065	20.7

of unit volume of " mixture " is constant to within about 2 per cent for all hydrocarbons with which we are concerned, and is only slightly lower for ethyl alcohol. The working fluid after combustion is approximately the same in each case—the proportions of CO_2 and H_2O alter, of course, but there is a large amount of inert nitrogen. The maximum power obtainable should therefore be closely proportional to the product of the heat per unit volume and the " volume-ratio " before and after combustion. The last column gives this product in each case—the extreme difference is about 3 per cent.

We show later that the actual difference between different hydrocarbons, when specific heats are allowed for accurately, is likely to be even less than this—a conclusion which is supported by some careful experiments made with a special single-cylinder engine of Mr. Ricardo's design.

It is a difficult matter to obtain strictly comparable results for maximum power, since this depends very noticeably on volumetric efficiency, or charge weight in cylinder, and this in its turn on the latent heat of evaporation of the fuels. Table II. gives figures which are the means of several observations for maximum power on four typical fuels.

TABLE II.

Fuel.	Max. I.M.E.P. observed.
Pure benzene	131.6
Petrol (all paraffins)	131.3
Pure cyclo-hexane	131.3
Pure ethyl alcohol	137.8

In these observations the same amount of heat was supplied (electrically) to the intake air in each case. The first three fuels, which represent the aromatic, paraffin, and naphthene groups respectively, show practically identical maximum power. The larger power with the ethyl alcohol is due to the very much greater relative magnitude of the latent heat to the heat of combustion: about 3.4 per cent as against 0.6 per cent to 1 per cent for the hydrocarbon fuels. The effect of the high latent heat is still more noticeable when running with methylated spirit, in which the latent heat is about 4.2 per cent of the heat of combustion; with this a maximum indicated mean effective pressure of 144.5 was obtained.

The cumulative evidence of a great number of tests on different types of spirit with this engine goes to show that there is no consistent difference in the maximum power obtainable with any hydrocarbon fuel, provided the conditions, *e.g.*, of heat supplied to the induction pipe for evaporation of the fuel, are the same in each case. We consider that all reports to the effect of greatly increased power caused by a change in fuel are due either to one fuel being on the point of "detonation," so that ignition has to be unduly retarded, or, in the case of multi-cylinder engines, to bad distribution in one case, but these are matters which we do not propose to consider in detail here.

III.

The specific heats and dissociation of carbon dioxide and water vapour are of the first importance in investigating the thermal efficiency of engines with different compression-ratios, working with different fuels and with different mixture strengths of the same fuel. It is well known that the power of an engine becomes a maximum when the mixture strength is slightly higher than that corresponding to complete chemical combustion to CO_2 and H_2O, and that if fuel in excess of this be used there is practically no diminution in power until the excess reaches some 30 per cent to 50 per cent. This behaviour has never yet been satisfactorily explained, but may be accounted for entirely when dissociation is taken properly into account. A method for calculating thermal efficiency from a knowledge of the specific heats was worked out very completely by the late Professor Hopkinson in his paper on the "Thermal Efficiency of Gas Engines," but all work on these lines in this country has suffered under two great disadvantages—

(1) The specific heats of the gases concerned were not known at all accurately above 1500° C.

(2) The dissociation of CO_2 and H_2O in the cylinder was unknown, and no attempt has been made to separate the two effects of the specific heat and of dissociation.

As to the first point, since the maximum temperature of the gases after explosion is of the order of 2500° C., and since the specific heats

both of CO_2 and of H_2O increase rapidly at high temperatures, it is clear that a knowledge of these up to 1500° C. only is quite inadequate. As to the second point, Sir Dugald Clerk pointed out some years ago that the experimental figures for "specific heats" derived from explosion experiments were "apparent specific heats," and would only be true specific heats if no dissociation had occurred in the experiments made to determine them.

The importance of this is twofold: first, since the dissociation of both CO_2 and H_2O is accompanied by an *increase* in the number of molecules, its extent at any temperature must depend upon the pressure, and hence the apparent specific heat must depend on the pressure; secondly, since both compounds have one dissociation product in common, namely, oxygen, dissociation in a mixture of the two is not the same as when they are separate. Thus at 2500° C., and atmospheric pressure, CO_2 is dissociated about 17 per cent, while water vapour is dissociated about 4 per cent, whereas in an equi-molecular mixture of the two in equilibrium at the same temperature and pressure the dissociation of the CO_2 is about 16 per cent, while that of water vapour is depressed to under 2 per cent, as follows from the ordinary law of mass action. The "apparent" specific heat of mixtures of CO_2 and water vapour, together with the other gases occurring in the cylinder after an explosion, may therefore be very different from the sum of the "apparent" specific heats of each measured separately. The only possible way to attack the problem is to find the *true* specific heats of the gases in question, and to measure the dissociation under definite conditions, from which it is possible to obtain sufficient data to calculate equilibrium conditions in any mixture.

Fortunately, sufficient data already exist for this purpose. A number of experiments, both on the dissociation of CO_2 and H_2O and on the true specific heats of the reacting gases, have been carried out in Nernst's laboratory in Berlin, with the primary object of testing recent developments in pure thermodynamics. The most important researches for our purposes are those of Pier and Bjerrum, a short critical account of which was given by one of the authors in *The Automobile Engineer* for February, 1920. It was there pointed out that for the purposes of calculating the efficiency of internal combustion engines the specific heats of all diatomic gases may be taken as the same as that of nitrogen, since the actual differences are slight, and the other diatomic gases are always present in small quantities. It was also pointed out that the measurements gave *true* specific heats, since precautions were taken, especially in the more recent experiments, to prevent dissociation by having an excess of one of the dissociation products present before the explosion.

The figures obtained are therefore true specific heats. They agree up to 1400° C. with the earlier figures of Holborn and Henning,

obtained by heating the gases at constant pressure. Above this temperature the new figures lie considerably below the old ones of Langen, as is to be expected, since the latter were "apparent" specific heats, and, moreover, Langen probably made insufficient allowance for heat-loss.

Table III., giving mean volumetric (or "molecular") heats, is quoted from the earlier paper. The actually observed figures were

TABLE III.—**Mean Volumetric Heats between 100° C. and T° C. in Gram Calories per Gram Molecule.**

100° C. up to.. ..	500	1000	1500	2000	2500	3000
Nitrogen	5.17	5.28	5.50	5.75	6.00	6.30
Water vapour ..	6.25	6.94	7.64	8.42	9.71	11.20
Carbon dioxide ..	8.25	9.55	10.07	10.50	10.87	10.95

made for nitrogen up to 2400° C., carbon dioxide up to 2700° C., and water vapour up to 3000° C. It seems probable that the figures for nitrogen and CO_2 are accurate to $\pm$ 4 per cent or less at 3000° C. Those for water vapour are probably not quite so reliable.

As regards dissociation, the same experimenters to whom we owe the recent figures for specific heats have, among others, fully investigated this for CO_2 and H_2O at high temperatures. Such dissociation depends upon pressure as well as temperature; in order to give an idea of how the degree of dissociation varies with temperature and

TABLE IV.

CO_2.

Temp.	Pressure 0.1	1.0	10.0	100.0 atmospheres
1500° C.	0.104	0.048	0.0224	0.01
2000° C.	4.35	2.05	0.96	0.445
2500° C.	33.5	17.6	8.63	4.09
3000° C.	77.1	54.8	32.2	16.9

H_2O.

1500° C.	0.043	0.02	0.009	0.004
2000° C.	1.25	0.58	0.27	0.125
2500° C.	8.84	4.21	1.98	0.927
3000° C.	28.4	14.4	7.04	3.33

pressure, the figures in Table IV. are quoted from Bjerrum's paper.* The degrees of dissociation are given as percentages.

* *Zeitschrift fur Physikalische Chemie*, 1912.

We have then, in the foregoing figures for true specific heat and for dissociation in terms of pressures and temperature, all the data necessary for the calculation of pressures and temperatures throughout the cycle of an engine in which the mixture strength and the chemical composition of the fuel are known.

IV.

In the subsequent calculations it is assumed that the only molecular species present after the explosion, and during the expansion, are CO_2, H_2O, CO, H_2, O_2, and N_2. The calculations show that the maximum temperature of explosion is of the order of 2500° C., and the temperature at the end of the expansion stroke about 2000° C. At these high temperatures all experiments go to show that the rate of chemical reaction is extremely high, and it is indeed assumed throughout this paper that the reaction velocity is such that chemical equilibrium is always maintained throughout the working stroke. A small departure from this in practice would not affect very much the calculated efficiencies, but any marked departure would lower the efficiency. Observed efficiencies are near enough to the calculated, when loss of heat is taken into account, to show that the assumption is substantially correct.

If equilibrium is constantly maintained during the working stroke, the pressure and temperature when the explosion takes place, and at any point during the expansion, can be calculated from the chemical equation representing the combustion, which may be written generally:

$$\begin{aligned} 2\,C_nH_m + \left(2n + \frac{m}{2}\right) O_2 \\ + 3.76\left(2n + \frac{m}{2}\right) N_2 \\ = (2n - x)\,CO_2 + x\,CO \\ + (m - y)\,H_2O + y\,H_2 \\ + 3.76\left(2n + \frac{m}{2}\right) N_2 \end{aligned}$$

and from the equations of dissociation of CO_2 and H_2O, namely,

$$\frac{(P_{CO})^2 \times P_{O_2}}{(P_{CO_2})^2} = K_1 \qquad \frac{(P_{H_2})^2 \times P_{O_2}}{(P_{H_2O})^2} = K_2$$

where P_{CO}, etc., represents the partial pressure of CO in the mixture, and K_1, K_2 are dependent on temperature only, and can be derived from the figures given in Table IV.

From the second law of thermodynamics, the change of the dissociation constant with the temperature can be given in the form

$$\frac{d \log_e K}{d T} = - \frac{Q}{R T^2}$$

where Q is the heat of dissociation per gram molecule, and the integration of this equation for not too wide temperature intervals is of the approximate form

$$\log_e K = \frac{Q}{R T} + \text{constant.}$$

All details of the approximate solutions of the equations are given in subsequent paragraphs, but it will be useful at this point to discuss some of the principal results.

The present work falls into three main groups :—

(*a*) An investigation into the degree of dissociation to be expected and its effect on the maximum temperatures reached in a petrol engine, and hence on the power output, by considering the specific case of an engine of 5 : 1 compression-ratio using pure benzene as fuel. Commercial benzole is rather more than 70 per cent benzene, so that the case is not far removed from practice.

(*b*) A comparison of maximum temperatures and power output at the same compression-ratio of 5 : 1 for three different fuels—benzene, C_6H_6 ; heptane, C_7H_{16} ; and alcohol, C_2H_6O—dissociation being allowed for. This will show what theoretical grounds there are, if any, for expecting a difference of power with different types of fuel. Heptane forms a considerable fraction of paraffin petrols ; its boiling point is 98.4° C., and it is taken as typical for the purposes of calculation.

(*c*) An examination of the effect on power and thermal efficiency of raising the compression-ratio when dissociation is allowed for.

The dissociation, explosion temperatures, etc., have been examined for a wide range of mixture strength, so that the variation of thermal efficiency with air/fuel ratio can be estimated. These figures can be directly compared with experimental results and a check thus obtained on the accuracy of the calculated conditions inside the cylinder. Maximum explosion temperatures have never been directly measured, but the calculated figures (2500-2700° C. for the richer mixtures, according to the fuel) are reasonable so far as experiments have gone when allowance is made for a certain heat-loss. The results of the work under headings (*a*), (*b*), and (*c*) above may be summarised conveniently as follows :—

(1) *The Effect of Mixture Strength.*—Pure benzene. The calculated results for explosion temperature and power output when dissociation is allowed for agree well with experimental results and show just that constancy of power over a wide range of mixture strength which

is found in practice. If dissociation is not allowed for, the calculated power is a maximum at the correct mixture for just complete combustion, and falls off steadily for both weaker and richer mixtures. The effect of dissociation is found to be to shift the maximum power point forward to a slightly stronger mixture, and to keep the power constant to within about 1 per cent for mixtures ranging from the "correct" to about 30 per cent richer than this.

(2) *Effect of Fuel.*—It is shown that, provided the initial temperature of the charge is the same (*i.e.*, the volumetric efficiency is constant), the power and thermal efficiency obtainable are the same to within 1 per cent for all types of hydrocarbon fuels, and also for ethyl alcohol. The results of experiments confirming the constancy of power have already been given, while the curve in Fig. 1 shows the

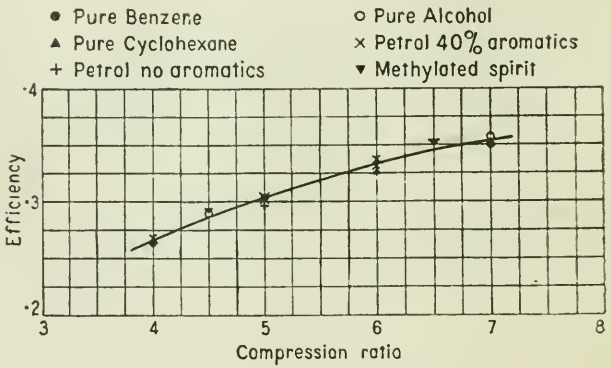


FIG. 1.

The above observations on thermal efficiency, while strictly comparable among themselves, are all slightly below the best obtained on the engine (and shown in Fig. 3), owing to all this series of observations having been made with only one of the two inlet valves in operation.

constancy of thermal efficiency observed from experiments at different compression-ratios with paraffin, aromatic and naphthene fuels, and alcohol. The experimental results thus confirm the theory completely.

(3) *Explosion Temperatures and Thermal Efficiencies.*—For a compression-ratio of 5 : 1, the calculated figures are as in Table V.

The effect of dissociation on the thermal efficiency for the "correct" mixture is to lower it by about 6 per cent below what would be obtained if there were no dissociation, and the products of combustion are given their true specific-heat values. For weak mixtures the dissociation is very much less, owing to the lower temperatures

reached ; at a 20 per cent weak mixture the explosion temperature is reduced by about 200 degrees C. by dissociation, *but the dissociation at the end of the expansion stroke is negligible, and the efficiency is not distinguishable from the "no dissociation" value.*

The calculated maximum thermal efficiency for the correct mixture is 33.8 per cent. This corresponds, at least very closely, to the

TABLE V.

Mixture.	Explosion temperature.	Thermal efficiency.
Correct—	° C.	Per cent.
If no dissociation	3023	0.359
Dissociation allowed for	2655	0.338
20 per cent weak—		
If no dissociation	2671	0.379
Dissociation allowed for	2470	0.379
50 per cent weak—		
If no dissociation	2000	0.412
Dissociation allowed for	1994	0.412

weakest mixture giving maximum power, but it is difficult to obtain accurate experimental figures with which to compare it, since the mixture strength can be varied so greatly, without altering the maximum power, by much more than corresponds to the errors of observation. On the other hand, the absolute maximum thermal efficiency in practice can be measured very accurately, since it occurs when "weak" mixtures are employed, and a considerable variation in mixture strength makes, in these circumstances, very little difference to the thermal efficiency.

In some tests by the late Professor W. Watson, using benzole as fuel, the maximum efficiency was obtained when running with an air/fuel ratio of 18.5 : 1 by weight ; this corresponds to a quantity of fuel about 0.8 of that at the "correct" mixture. Our own experiments lead us to the same conclusion, and therefore, in order to get a true comparison between theory and experiment, and a correct estimate of the margin of heat-loss to the cylinder, the theoretical efficiency with which the best experimental figures should be compared is that of an engine working on a 20 per cent weak mixture.

We find this to be very nearly 38 per cent at a compression-ratio of 5 : 1, whereas the most reliable experimental figure for a maximum indicated thermal efficiency at this compression-ratio which the authors have been able to obtain is just under about 32 per cent at 1500 revs. per minute. The margin left for heat-loss to the cylinder during explosion and expansion is about 13 per cent of the total heat (see Appendix V., p. 47). Since a large part of the heat-loss to the cooling water will take place during the exhaust stroke, and round

the exhaust-valve seating when the gases are leaving the cylinder, this figure of 13 per cent on expansion, when compared with a measured *total* jacket loss of about 25 per cent, is of the right order. Until more exact data are available as to mixture strength at the point of maximum economy, it can only be said that the calculated thermal efficiencies are reasonable, and afford a satisfactory check to the correctness of the assumptions on which the theory has been worked out.

If our results are admitted to be substantially correct, the conclusion to be drawn is that the indicated thermal efficiencies of the best engines are not likely to be improved upon substantially by any development of design other than that of the employment of weaker mixtures than can now be used, or by an increase in the expansion-ratio. Our results show that great improvement is to be expected if either of these methods is used. The effect of increasing the expansion-ratio is dealt with below; Fig. 2 shows

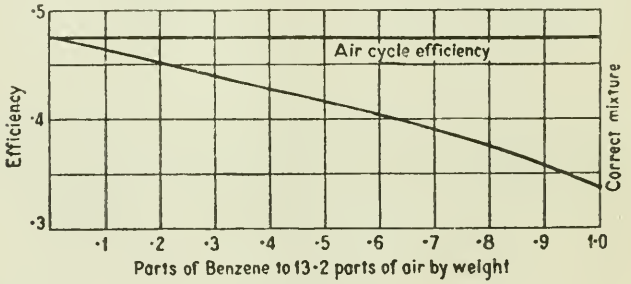


FIG. 2.
Maximum thermal efficiencies at 5 : 1 expansion-ratio.
Benzene and air mixtures of different strengths.

the possibility of increasing thermal efficiency by the use of weak mixtures. The thermal efficiency of a 50 per cent weak mixture at a 5 : 1 expansion-ratio is calculated to be 41.2 per cent, while for the limiting case of zero fuel the calculated efficiency would coincide with the "air standard"* efficiency. Such an efficiency as that with 50 per cent weak mixture could not be approached except by means of the injection of liquid fuel into the cylinder, as is done in Diesel engines, or by the use of some such device as Ricardo's "stratified charge." The reason for this is that a *homogeneous* mixture of fuel vapour and air cannot be exploded if the concentration of the fuel vapour is appreciably less than 80 per cent of that required for complete combustion of the air. If weaker mixtures are to be exploded, the mixture must be heterogeneous at the moment of explosion, or the initial temperature must be increased considerably.

* NOTE.—The term "air standard" is used throughout in the text in preference to the term "air cycle."

(4) *Effect of Increasing Expansion-ratio* (*see Fig.* 3).—We show that the relative increase in thermal efficiency to be expected from an increase in the expansion-ratio is distinctly higher than the "air standard" efficiencies would suggest. As already stated, the theoretical efficiency for the correct mixture at 5 : 1 compression is 33.8 per cent or about 71 per cent of the air standard efficiency.

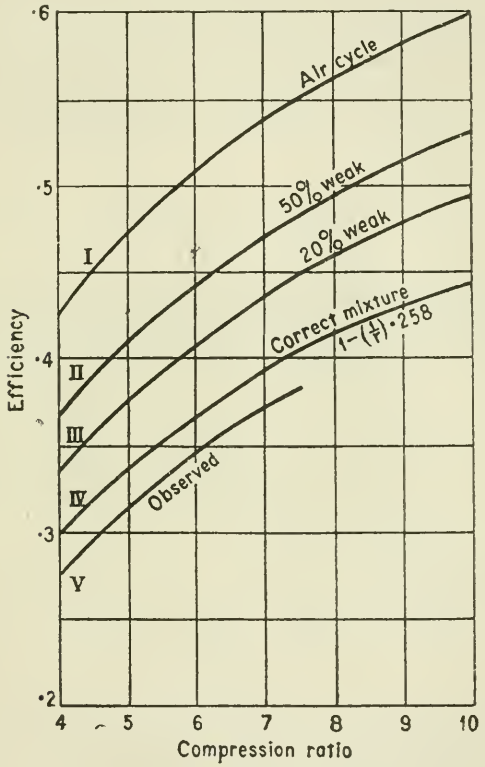


Fig. 3.
Effect of increasing expansion-ratio.

The theoretical efficiency for a 10 : 1 compression is 45 per cent, or nearly 75 per cent of the air standard efficiency. Our results for any compression-ratio between 4 : 1 and 10 : 1 can be closely expressed by the following equations :—

For "correct" mixtures :

$$\text{Maximum theoretical efficiency} = 1 - \left(\frac{1}{r}\right)^{0.258}$$

For 20 per cent weak mixtures :

$$\text{Theoretical efficiency} = 1 - \left(\frac{1}{r}\right)^{0.296}$$

These equations are empirical in the sense that it must not be assumed that, *e.g.*, the average value for γ for the products of combustion of a 20 per cent weak mixture is 1.296, but the form in which they are expressed, corresponding as it does to the air standard formula, makes their use convenient, and the results to which they lead represent with sufficient accuracy the figures obtained from the theoretical calculations. As already stated, the 20 per cent weak mixture corresponds very closely to the conditions under which the maximum observed efficiencies are now obtained, and we therefore propose the equation

$$\text{Maximum efficiency} = 1 - \left(\frac{1}{r}\right)^{0.296}$$

to replace the present air standard formula.

The calculated relative increase in efficiency on raising the compression-ratio was unexpected, and experiments were immediately carried out to see if the result would be confirmed. Fortunately

TABLE VI.

Compression-ratio.	Max. thermal effic'y.		Ratio. Observed / Calculated.	Air standard efficiency.	Ratio. Observed / Air standard.
	Calculated from equation.	Observed.			
4 to 1	0.337	0.277	0.82	0.425	0.65
5 to 1	0.380	0.316	0.83	0.475	0.665
7 to 1	0.440	0.372	0.85	0.540	0.685

we were able to make use of an experimental engine designed by Mr. Ricardo for this purpose, in which compression-ratio could be varied while running without affecting any other conditions.

It is clear from Table VI. that the calculated rise of relative thermal efficiency is fully confirmed. It is thought that this important result has not previously been suspected ; in fact, some writers have expressed a belief that thermal efficiencies would not be improved by raising the compression so much as indicated by the air standard formula.

V.

As already stated, it has been assumed throughout the calculations that no gases other than CO_2, H_2O, CO, H_2, O_2, and N_2 are present in

the cylinder during the working stroke. It is, however, possible that other gases do exist. For example, methane is known to result from the interaction of carbon monoxide and hydrogen at moderately low temperatures, and has often been shown to exist in exhaust gases; further, there is the possibility of the formation of oxides of nitrogen.

The presence of any such compounds in the equilibrium mixture in the cylinder after explosion would tend, if anything, to lower the thermal efficiency, and not to raise it, since the "apparent" specific heat of the mixture would be increased; but, at the same time, it is theoretically possible for perceptible amounts to be present just after explosion without materially affecting the efficiency, provided that the amounts present at the end of the expansion stroke were negligible.

The theoretical results for the efficiency of an engine running on a 20 per cent weak mixture at a 5 : 1 compression may be quoted to illustrate this point. The calculated dissociation of CO_2 at the maximum temperature of explosion in this case is about 20 per cent, but at the lower temperature at the *end* of the expansion stroke the dissociation is negligible. The work done by the expansion in such a case is practically the same as if no dissociation at all occurred; in fact, the only effect of dissociation is to lower the maximum temperatures and pressures without materially altering the efficiency. It is therefore of considerable interest to examine the possibility of the formation of other compounds.

The formation of nitric oxide is endothermic.

$$2\,NO = N_2 + O_2 + 43{,}200 \text{ calories},$$

and the proportion present at equilibrium will therefore tend to increase with the temperature. The reaction constant is given by the equation

$$\frac{(P_{NO})^2}{P_{N_2} \times P_{O_2}} = K.$$

The formation of nitric oxide from air has been the subject of much experiment, the most reliable data for the equilibrium mixture at the temperatures we are considering being probably those of Nernst* who gives the following results, x being the proportion by volume of NO formed :—

T (abs.) observed.	x per cent observed.
1811	0.37
2033	0.64
2580	2.05

* *Zeitschr. f. anorgan. Chem.* 49, 213, 1906.

From these figures it can be shown that log K is very closely given, for this range of temperature by the equations

$$\log K = -\frac{9450}{T} + 1.12$$

The proportion of nitrogen to oxygen at the maximum temperatures of explosion of benzene and air are given in Table VII. (page 22), from which it may be calculated from the above equation that the amount of nitric oxide in the cylinder is only about 2 per cent of the nitrogen present at the explosion temperature. This calculation is for the correct mixture; for a rich mixture, since the oxygen pressure is greatly diminished, that of NO will also diminish, whereas it will increase for a weak mixture.

At the end of the expansion stroke, if equilibrium is maintained, only a negligible amount of nitric oxide can be present.

The formation of methane from carbon monoxide and hydrogen according to the equation

$$CO + 3\,H_2 = CH_4 + H_2O$$

has apparently only been studied at low temperatures, but a sufficiently good idea of the amount likely to be formed at high temperature can be obtained by Nernst's approximate formula. This takes the form

$$\log K = \log \frac{P_{CO} \times P^3_{H_2}}{P_{CH_4} \times P_{H_2O}} = -\frac{19000}{T} + 3.5 \log T + 2.2$$

whence K = about 10^{12} at 3000° absolute and 10^8 at 2000° absolute, from which it is clear that no methane could possibly be detected during the expansion stroke. On the other hand, when T = 1000, K is of the order of 50, and hence methane appears in the exhaust, and is, in fact, probably only formed after the gases have left the cylinder.

Finally, there is the question of formaldehyde, which may be formed from CO and H_2.

$$CO + H_2 = H\,.\,CHO.$$

To obtain an approximate idea of the possibility of this reaction taking place it is necessary to know the heat of formation, or alternatively the heat of combustion of formaldehyde. Practical difficulties are too great for this to be determined with any accuracy. The lower heat of combustion of acetaldehyde, CH_3CHO, is 262 kg. calories per gr. molecule (Thomsen), and that of propionaldehyde is 412, an increase of 150 kg. calories for an increase of molecular weight of 14 (see Section I.). The lower heat of combustion of H . CHO must therefore be approximately 110 kg. calories. That of CO is 68 and of H_2 is 59 kg. calories, and hence the molecular heat of formaldehyde from CO and hydrogen must be of the order of (68 + 59 − 110) kg. calories = 17,000 calories.

Too much reliance cannot be placed on this figure, but it may be accepted that the formation takes place with evolution of heat, and that therefore a high temperature will tend to prevent—not to promote—the formation of aldehydes. This is contrary to the statements of some investigators.

The value of the equilibrium constant

$$\frac{P_{CO} \times P_{H_2}}{P_{H\,.\,CHO.}} = K$$

may again be estimated at different temperatures from Nernst's approximation formula

$$\log K = -\frac{Q}{4.57\,T} + 1.75 \log T + C$$

or in this case

$$\log K = -\frac{3700}{T} + 1.75 \log T + 2.1$$

If $T = 3000°$ $K = 10^7$ approximately
If $T = 2000°$ $K = 10^5$ approximately
If $T = 1000°$ $K = 10^4$ approximately

so that at no period during the cycle should any appreciable amount of formaldehyde, and similarly of other aldehydes, be present in equilibrium with the other gaseous constituents.

It seems, therefore, that no other compounds, other than nitric oxide, if the mixture is weak, are likely to be present in the cylinder during the working stroke in appreciable proportions, and that in any case their presence cannot affect appreciably the theoretical thermal efficiency of the engine.

General thermodynamical considerations of this kind lead to conclusions of some interest with regard to the composition of exhaust gases. Exhaust gases leave the cylinder at a fairly high temperature, probably well above 1000° C., and are then quickly cooled to a temperature at which rate of chemical reaction is very slow. We should therefore expect the composition of the exhaust gases to correspond approximately to equilibrium conditions at, say, 1500° absolute. The proportions of carbon monoxide should therefore be calculable with some degree of accuracy from a knowledge of the equilibrium constant of the water gas reaction

$$CO + H_2O = CO_2 + H_2$$

at this temperature.

The equilibrium constant

$$K = \frac{P_{CO} \times P_{H_2O}}{P_{CO_2} \times P_{H_2}}$$

has been measured by Hahn* and has a value of about 3 at 1500° absolute.

* *Zeitschr. f. Physick. Chem.*, 44, 513, and 48, 735.

Hence the ratio of hydrogen to carbon monoxide in the exhaust must be about one-third of the ratio of water vapour to carbon dioxide, or one-sixth of the ratio of hydrogen atoms to carbon atoms in the fuel.

If the fuel is a petrol consisting mainly of paraffin hydrocarbons, the atomic ratio H/C will be approximately 2.3 (heptane), and therefore the ratio H_2/CO in the exhaust will be approximately 0.4.

This corresponds almost exactly to an empirical rule known as Ballantyne's rule, which states that the ratio H_2/CO in the exhaust gases = 0.36*, but it is clear that this explanation shows that Ballantyne's "rule" must be applied with caution, since, if the fuel were, for example, benzene, with a ratio H/C = 1, the ratio H_2/CO in the exhaust would be more nearly = 0.17.

A similar calculation can be made on the percentage of methane in the exhaust, but the case is somewhat more complicated, and the estimation loses accuracy from the lack of accurate experiments on the reaction.

Part II.

I.

Before dealing in detail with the question of dissociation it will be as well to make clear certain preliminary conditions, and to give figures, useful later for comparison, calculated on the assumption that no dissociation takes place.

Before compression begins, the mixture of fuel and air is assumed to be at 100° C. and atmospheric pressure. The work done on the mixture during compression will depend on its mean specific heat, and it will simplify calculations if we take this as being the same as the mean specific heat of the mixture after combustion for the same temperature range. The difference is small and the effect on the compression work quite negligible. The point is dealt with in detail in Appendix I., from which it will also be clear that it is not necessary to know accurately the specific heat of the fuel vapour.

The work done in compressing the mixture has been taken to be :† for compression—

5 : 1 . . 1.59 kilo. cal. per gram-mol.‡
10 : 1 . . 2.59 kilo. cal. per gram-mol.‡

The small differences of specific heat for different mixture strengths are negligible in this connection, and the above values have been adopted throughout.

* See Dugald Clerk, p. 627, 1913 edition.
† For calculation, see Appendix I., p. 43.
‡ One kilo. cal. per gram-molecule = 3960 ft.-lb. per standard cubic foot.

The heats of combustion of the fuels used are accurately known from calorimetric measurements. They are :—

Benzene, 751 kilo. calories per gram-molecule (lower value).
Heptane, 1061 kilo. calories per gram-molecule (lower value).
Alcohol, 308 kilo. calories per gram-molecule (lower value).
Hydrogen, 58.7 kilo. calories per gram-molecule (lower value).
Carbon momoxide, 68.0 kilo. calories per gram-molecule.

There is some slight uncertainty as to heptane and alcohol, but even here probably less than 1 per cent.

All these heats of combustion are, of course, measured under calorimetric conditions, *i.e.*, starting at room temperature of about 20° C. The error involved is unavoidable, but is probably not large, since

$$\frac{d\ \mathrm{H}}{d\ t} = \mathrm{C_1} - \mathrm{C_2}$$

where H is the heat of combustion and C_1 and C_2 are the thermal capacities of the mixture before and after combustion, and the difference between these, as appears from Appendix I., is small.

II.

Calculations when No Dissociation is Assumed.

In these circumstances the combustion of a normal or " correct " mixture of benzene and air will take place according to the equation—

$$2\ \mathrm{C_6H_6} + 15\ (\mathrm{O_2}) + (56.4\ \mathrm{N_2}) = 12\ \mathrm{CO_2} + 6\ \mathrm{H_2O} + (56.4\ \mathrm{N.})$$

Number of molecules before combustion = 73.4.
Number of molecules after combustion = 74.4.
Heat of combustion of 1 gram-molecule of mixture

$$= \frac{2 \times 751}{73.4} = 20.46 \text{ kg. cal.} = \mathrm{Q\ (say)}.$$

If q = energy added on compression, and E_T = the " energy content " of the products of combustion at any absolute temperature T (the energy at 100° C. being arbitrarily taken as zero), then after combustion

$$E_T = Q + q = 20.46 + 1.59 = 22.05 \text{ kg. cal. per gram-mol. of } \textit{original} \text{ mixture.}$$

The mean volumetric heat of the products of combustion may be calculated from a knowledge of the mean volumetric heats of N_2, CO_2, and H_2O. If one gram-molecule of mixture is taken as the standard volume, the energy content of the products of combustion at T° is C × (T − 373), where C is the mean volumetric heat between 373° absolute and T.

Calculation for the above mixture gives the figures, in kg. calories per molecule of original mixture :—

$t°$ C.	E.
100	0
500	2.33
1000	5.56
1500	9.09
2000	12.96
2500	17.22
3000	21.85

" E " can be very nearly expressed by an equation of the form

$$E = a\,(T - 373) + b\,(T - 373)^2$$

for the range of temperature with which we are concerned. It is not necessary to try and express the whole range by an equation of this form, since the range of temperature, which is of importance for our purpose, is that between about 2000° C. and 3000° C.

Calculation of Maximum Temperature.

The maximum temperature T_e on explosion (assuming no dissociation) can now be calculated from the " energy curve " of the products of combustion of one mol. of mixture, since the total energy put into the gas to raise its temperature from 373 absolute to $T_e = 1.59 + 20.46 = 22.05$ kg. calories.

The energy curve from 2000° to 3000° C. can be very closely represented by the equation—

$$E_T = 0.549\,\frac{(T - 373)}{100} + 0.00703\,\frac{(T - 373)^2}{100}$$

$t°$ C.	T (abs.)	E_T calc. from equation.	E_T from specific heat results.
2000	2273	12.97	12.96
2500	2773	17.23	17.22
3000	3273	21.83	21.85

Hence T_e is given by—

$$22.05 = 0.549\,\frac{(T_e - 373)}{100} + 0.00703\,\frac{(T_e - 373)^2}{100}$$

$$T_e = 3296 \text{ abs.} = 3023° \text{ C.}$$

Calculation of the Work Done on Expansion.

Calculation of the work done on expansion requires strictly a knowledge of the variation of true (not mean) specific heat with the temperature. This can be obtained from the equations given above, for

$$C_T = \frac{d\,E}{d\,T} = 4.965 + 0.001406\,T$$

over the range of temperature under consideration.

The final temperature T_f is then calculated from the equation

$$\int p\,dv = \int c\,dT$$

or $$R \ln 5 = 4.965 \ln \frac{T_e}{T_f} + 0.001406 (T_e - T_f),$$

where R is the gas constant, which, however, refers not to one mol. of product, but to $\frac{74.4}{73.4}$ mols., and therefore

$$= 1.985 \times \frac{74.4}{73.4} = 2.012.$$

Calculation of T_f by such a method is, however, laborious, and it leads to little loss of accuracy* if a mean constant value of C over the range T_f to T_e is assumed, when the equation reduces to

$$\frac{T_e}{T_f} = 5^{\gamma - 1} \quad \ldots\ldots\ldots\ldots (1)$$

where "γ" is the mean value, corresponding to the mean value of "C," which is also given by the relation

$$C = \frac{E_{T_e} - E_{T_f}}{T - T_f} \quad \ldots\ldots\ldots\ldots (2)$$

The calculation is made by "guessing" a value for γ, calculating T_f from equation (1), substituting in equation (2), and hence obtaining a value for the mean specific heat which should agree with the value for γ originally taken.

For example, let $\gamma = 1.224$.

This gives

$$\frac{T_e}{T_f} = 5^{0.224} = 1.434$$

or $$T_f = \frac{3296}{1.434} = 2298 = 2025^\circ \text{ C.}$$

$$E_{T_f} = 0.549 \times 19.25 + 0.00703 (19.25)^2$$
$$= 10.57 + 2.605 = 13.175$$

$$C_{T_e \cdot T_f} = \frac{22.05 - 13.17}{3296 - 2298} = \frac{8.87}{998} = 8.89$$

$$\gamma = \frac{8.89 + 2.01}{8.89} = 1.226.$$

If $\gamma = 1.225$:

$$\frac{T_e}{T_f} = 5^{0.225} = 1.437 \quad T_f = \frac{3296}{1.437} = 2294 = 2021^\circ \text{ C.}$$

* For calculation of the error involved, see Appendix II., p. 44.

$$E_{T_f} = 0.549 \times 19.21 + 0.00703\,(19.21)^2$$
$$= 10.545 + 2.594 = 13.14$$
$$C_{T_e T_f} = \frac{22.05 - 13.14}{3296 - 2294} = \frac{8.91}{1002} = 8.89$$

$\gamma = 1.226$ as before.

If $\gamma = 1.226$:

$$\frac{T_e}{T_f} = 5^{0.226} = 1.439 \quad T_f = \frac{3296}{1.439} = 2290 = 2017^\circ \text{ C.}$$
$$E_{T_f} = 0.549 \times 19.17 + 0.00703\,(19.17)^2$$
$$= 10.525 + 2.583 = 13.108$$
$$C_{T_e T_f} = \frac{22.05 - 13.11}{3296 - 2290} = \frac{8.94}{1006} = 8.89$$

$\gamma = 1.226$ as before.

∴ Final temperature = 2017° C.

Work done on expansion = 8.94 kg. cals. per mol. mixture. The net work done during the cycle is

8.94 − compression work = 8.94 − 1.59 = 7.35 kg. cals.

Table VII.

Mixture	Composition of products of combustion.						Vol. V_1 before combustion.	Vol. V_2 after combustion.	Ratio $\frac{V_2}{V_1}$	Heat of comb. per mol. original mixt. +1.59.	Max. temp. of explosion. ° C.
	N_2	CO_2	CO	H_2O	H_2	O_2					
50% we'k	56.4	6.0	—	3.0	—	7.5	72.4	72.9	1.005	11.96	2000
20% we'k	56.4	9.6	—	4.8	—	3.0	73.0	73.8	1.011	18.05	2671
Correct	—	12.0	—	6.0	—	—	73.4	74.4	1.013	22.05	3023
10% rich	56.4	10.30	2.90	6.50	.10	—	73.6	76.2	1.035	21.28	2927
20% rich	56.4	8.71	5.69	6.89	.31	—	73.8	78.0	1.056	20.52	2835
50% rich	56.4	4.67	13.33	7.33	1.67	—	74.4	83.4	1.120	18.37	2569

The efficiency is $\frac{7.35}{20.45} = 35.9$ per cent.

The "air-standard" efficiency $1 - \left(\frac{1}{r}\right)^{\gamma-1}$

$= 47.5$ per cent ($\gamma = 1.4$); 35.9 per cent corresponds to a mean value for γ of 1.275.

Table VII. shows the maximum temperatures reached, and the composition of the products of combustion, for a series of mixtures of benzene and air both weaker and richer than the "correct" mixture. No dissociation is assumed, but when the mixture is rich the equilibrium proportions of CO_2, CO, H_2O, and H_2 have been calculated by the method given in what follows.

III.

Calculation of Maximum Temperatures Allowing for Dissociation.

Owing to dissociation, the maximum temperatures calculated in Table VII. are not reached, and the efficiency of the process has been over-estimated.

Sufficient data on the degree of dissociation of CO_2 and H_2O at different temperatures now exist to enable the effect of the maximum temperature on explosion, and on the efficiency, to be estimated. Figures for the dissociation of CO_2 and H_2O when *separate* at various temperatures and pressures were given in Table IV., p. 7. The heats of combustion of CO and H_2 to CO_2 and H_2O were also given as 68.0 and 58.7 kg. calories per gm.-mol. respectively.

The calculation of maximum temperatures when dissociation is taken into account is a matter of trial and error. The heat liberated depends on the amount of dissociation, which itself depends on the final temperature reached.

The first question of importance is the amount of dissociation if we start with the "correct" mixture. If no dissociation occurred, the combustion of this would take place according to the equation :

$$2\,C_6H_6 + 15\,O_2 + (56.4\,N_2) = 12\,CO_2 + 6\,H_2O + (56.4\,N_2).$$

If dissociation of the CO_2 and H_2O occurs we may write the equation

$$2\,C_6H_6 + 15\,O_2 + (56.4\,N_2) = A.\,CO_2 + B.\,CO + C.\,H_2O + D.\,H_2 + E.\,O_2 + (56.4\,N_2)$$

In order to find the quantities A, B, C, D, and E we need five independent relations connecting them. We can write down at once :

$$A + B = 12$$
$$2\,A + B + C + 2\,E = 30$$
$$2\,C + 2\,D = 12$$

For the necessary further equations we must go to equilibrium conditions based on experimental data on dissociation. The figures given in Table IV. represent the dissociation of each gas separately when there is no excess of the products of dissociation. Any excess of such products will diminish the amount of dissociation by an amount which can be calculated from the fundamental equations :

$$\frac{(P_{CO})^2 \times P_{O_2}}{(P_{CO_2})^2} = K_1 \quad \frac{(P_{H_2})^2 \times P_{O_2}}{(P_{H_2O})^2} = K_2$$

where K_1 and K_2 depend on the temperature only, and P_{CO_2}, etc., represent the partial pressures of CO_2, etc., in the equilibrium mixture.

Moreover, the reaction

$$CO_2 + H_2 = CO + H_2O$$

has been studied at different temperatures, and it has been shown that an equilibrium exists which can be represented with sufficient accuracy for our purposes by the equation

$$\log. K = \alpha - \frac{\beta}{T},$$

where T is the absolute temperature and

$$K = \frac{CO \times H_2O}{CO_2 \times H_2}.$$

The dependence of K_1 and K_2 upon the temperature can be expressed in a form similar to K. It is unnecessary, however, to use the expression for K_2, since a knowledge of K_1 and the equilibrium

$$\frac{CO \times H_2O}{CO_2 \times H_2}$$

gives all that is required. [K is connected to K_1 and K_2 by the equation $K = \sqrt{K_1/K_2}$.]

Since partial pressures are proportional to the number of molecules of each gas present in the mixture, we see that, in addition to the three equations already given connecting the coefficients A, B, C, D and E, the two expressions

$$\frac{B \,.\, C}{A \,.\, D} \text{ and } \frac{B^2}{A^2} E$$

are quantities which depend, the first on the temperature only, and the second on the temperature and pressure.

As regards the dependence of K_1 and K upon the temperature, suppose

$$\log K_1 = \alpha^1 - \frac{\beta^1}{T}$$

and

$$\log K = \alpha - \frac{\beta}{T}.$$

Then we have, according to the results of Nernst and others :

For K_1 at 2000° .. .. .. 4.49×10^{-6}
,, 2500° .. .. .. 4.01×10^{-3}
,, 3000° .. .. .. 3.94×10^{-1}

The variation between these limits is closely given by taking the following values of the constants α^1 and β^1 :

	α^1.	β^1.
2000°—2500°	9.41	29500
2500°—3000°	9.57	29920

For α and β, according to the results of Nernst, the following figures may be taken :—

	α.	β.
2000°—2500°	1.70	1750
2500°—3000°	1.916	2290

Corresponding values for K are given in Table VIII.

TABLE VIII.

Temp. abs. °C.	K.	Temp. abs. °C.	K.
2000	6.7	2600	10.7
2100	7.3	2700	11.6
2200	8.0	2800	12.4
2300	8.7	2900	13.2
2400	9.3	3000	14.0
2500	10.0		

The calculation of the maximum temperature is carried out by approximation. The temperature is "guessed;" this fixes the products $\frac{B^2}{A^2}E$ and $\frac{BC}{AD}$; A, B, C, D, and E can then be calculated, since we have five equations connecting them. From these values the heat liberated on combustion and the specific heats of the products of combustion can be calculated, and from a knowledge of these two a maximum temperature on explosion is found. This should agree with the temperature guessed in the first place. If it does not, a second value is taken nearer the truth. The calculation is laborious, but quite straightforward, and may best be illustrated by taking the particular case of the correct benzene mixture.

For this, as shown previously, the number of molecules before combustion is 73.4, and we have the equations :—

$$A + B = 12 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$
$$2A + B + C + 2E = 30 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$
$$2C + 2D = 12 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Now, if, after explosion,

T_e = maximum temperature
P_e = maximum pressure
X = total number of molecules
$(= A + B + C + D + E + 56.4)$

then

$$P_{O_2} = \frac{E}{X} \cdot P_e \text{ and } P_e = \frac{X}{73.4} \times 5 \times \frac{T_e}{373}.$$

Now, if T_e is guessed at 2500° C. = 2773 abs.

then $K_1 = 0.06$

and $K = 12.2$

whence $$P_{O_2} = \frac{E}{73.4} 5 \cdot \frac{2773}{373} = 0.506\ E$$

and $$\frac{B^2}{A^2} E = \frac{K_1}{0.506} = 0.119 \quad \ldots\ldots \quad (4)$$

$$\frac{B \cdot C}{A \cdot D} = 12.2 \quad \ldots\ldots\ldots \quad (5)$$

From equations 1–5 we get :

A = 9.3
B = 2.7
C = 5.86
D = 0.14
E = 1.42.

The corresponding heat of combustion and calculated maximum temperature are shown in the first line of Table IX.*

TABLE IX.

Guessed maximum temperature.	Composition of products of combustion.					Heat of combust'n per mol. mixture + 1.59 kilo. cal.	Therm. capacity of products per mol. mixture.	Calculated max. temperature, ° C.
	CO_2.	CO.	H_2O.	H_2.	O_2.			
2500	9.30	2.70	5.86	0.14	1.42	19.44	7.11	2835
2550	8.94	3.06	5.84	0.16	1.61	19.10	7.14	2775
2600	8.58	3.42	5.82	0.18	1.81	18.75	7.17	2715
2650	8.22	3.78	5.80	0.20	1.99	18.39	7.20	2655
2700	7.86	4.14	5.78	0.22	2.18	18.04	7.22	2599
2750	7.50	4.50	5.76	0.24	2.37	17.69	7.25	2541

* For details of these calculations, see Appendix III., p. 44.

It will be seen that the temperature found (2835° C.) is much above the guessed value. Successive approximations are shown in the Table, and it will be seen that when the temperature guessed is 2650, the calculated result is 2655, so that the latter may be taken as the correct maximum temperature (assuming instantaneous combustion and no heat-loss), and the true equation representing the process of combustion is :

$$2\,C_6H_6 + 15\,O_2 + 56.4\,N_2 = 8.22\,CO_2 + 3.78\,CO + 5.8\,H_2O + 0.20\,H_2 + 1.99\,O_2 + 56.4\,N_2.$$

To summarise the results, therefore :—If a correct mixture of benzene vapour and air at 100° C., and an initial pressure of one atmosphere, is compressed to one-fifth of its volume and exploded, the carbon will not be completely burned to CO_2, but 31.5 per cent of it only will be burnt to CO. The total heat added to the mixture by compression and combustion is 18.39 kilo. calories, as against 22.05 if there had been no dissociation, and the maximum temperature reached will be 2655° C. instead of 3023° C.

The number of molecules after combustion is 76.39, say, 76.4, and the ratio of volume increase due to combustion is $\frac{76.4}{73.4} = 1.041$.

The maximum pressure on explosion will be :

$14.7 \times 5 \times 1.041 \times \frac{2928}{373} = 600.7$ lb. per sq. in., as against

$14.7 \times 5 \times 1.013 \times \frac{3296}{373} = 658$ if there had been no dissociation.

IV.

Maximum Temperatures of Rich Mixtures.

All the foregoing calculations have been made for a correct mixture of benzene and air, although a Table was given at the end of Section II., p. 22, showing maximum temperatures for rich mixtures on the assumption of no dissociation.

The effects of dissociation are particularly interesting when rich mixtures are considered, for it will appear that when dissociation is allowed for the maximum temperature is *not* reached when the mixture is "correct," as is the case in Table VII., but when it is some 20 per cent richer than this. Moreover, over a considerable range of mixture strength there is only a small variation in the maximum temperature, and, therefore, also of maximum pressure. Now, speaking generally, this corresponds closely to what is found in practice, since the power of a petrol engine varies only slightly over a wide range of mixture strength.

If the mixture before compression is rich in benzene, there will be more CO present after explosion, and this will depress the dissocia-

TABLE X.—**Compression-Ratio 5 : 1.**

Deg. of richness a.	Composition of products of combustion. N_2.	CO_2.	CO.	H_2O	H_2.	O_2.	Vol. before combustion.	Vol. after combustion.	Vol. ratio k.	Mean vol. heat per mol. mixt.	Heat of combustion + 1.59.	Max. temp. ° C.
0.5	56.4	5.95	0.05	3.00	0.00	7.52	72.4	72.9	1.007	6.30	11.93	1994
0.2	56.4	7.70	1.90	4.70	0.10	4.00	73.0	74.8	1.025	6.86	16.23	2470
0	56.4	8.22	3.78	5.80	0.20	1.99	73.4	76.4	1.040	7.20	18.39	2655
0.1	56.4	7.93	5.27	6.31	0.29	1.28	73.6	77.5	1.052	7.31	18.93	2690
0.2	56.4	7.46	6.94	6.74	0.46	0.70	73.8	78.7	1.065	7.40	19.24	2701
0.3	56.4	6.69	8.91	7.10	0.70	0.305	74.0	80.1	1.081	7.45	19.22	2681
0.4	56.4	5.69	11.11	7.31	1.09	0.10	74.2	81.7	1.100	7.46	18.87	2630
0.5	56.4	4.60	13.40	7.36	1.64	0.02	74.4	83.4	1.120	7.45	18.32	2561

tion of CO_2 and lower the partial pressure of oxygen. In fact, it is obvious that with a very rich mixture practically all the carbon will

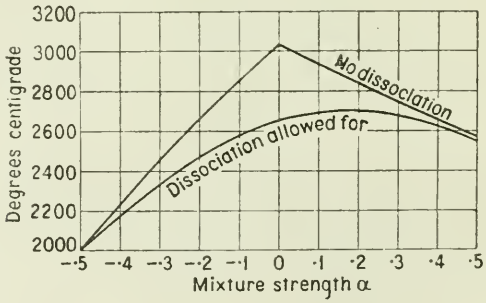


FIG. 4.

Explosion temperatures. Benzene.

be burnt to CO, and the amount of oxygen will be negligible. The calculation of the proportions of the various gases present is carried out in exactly the same way as was given for the correct mixture.

If a is the degree of richness, we have as the combustible mixture

$$2\,(1 + a)\,C_6H_6 + 15\,O_2 + 56.4\,N_2.$$

The equations for solution are :

$$A + B = 12\,(1 + a)$$
$$2\,A + B + C + 2\,E = 30$$
$$C + D = 6\,(1 + a).$$

$\frac{B^2}{A^2} \times P_{O_2}$ and $\frac{B \,.\, C}{A \,.\, D}$ are, as before, quantities which are fixed when the temperature is known.

The results for mixtures of benzene and air are given in Table X. and are shown plotted in Fig. 4 by the side of the calculated temperatures for the same mixtures, previously given in Table VII., assuming that no dissociation takes place.

V.

Maximum Temperatures with Heptane and Alcohol as Fuel.

Before dealing with the pressures and temperatures on the expansion stroke, and hence the work done, it will be as well to give at this point the result of calculations upon heptane and alcohol similar to those just given for benzene. These results are shown in Tables

Table XI.—**Heptane.**

Deg. of richness a.	Composition of products of combustion.						Vol. before combustion.	Vol. after combustion.	Vol. ratio k.	Mean vol. heat per mol. mixt.	Heat of comb. per gm. mol. +1.59.	Max. temp. ° C.
	N_2	CO_2	CO	H_2O	H_2	O_2						
0	82.8	10.27	3.73	15.53	0.47	2.10	106.8	115.9	1.085	7.56	18.83	2591
0.1	82.8	9.74	5.66	16.86	0.74	1.00	107.0	116.8	1.091	7.69	19.40	2623
0.2	82.8	8.64	8.16	17.92	1.28	0.32	107.2	119.1	1.111	7.78	19.47	2603
0.3	82.8	7.14	11.06	18.52	2.28	0.07	107.4	121.9	1.135	7.84	19.03	2527
0.4	82.8	5.71	13.89	18.66	3.74	0.013	107.6	124.8	1.159	7.83	18.38	2448
0.5	82.8	4.65	16.35	18.35	5.65	0.00	107.8	127.8	1.185	7.82	17.72	2366

XI. and XII. In Section VI. a general discussion will be given of the comparative results on the three fuels as exhibited in Tables VII., X., XI., and XII.

The general equations for the combustion of mixtures of heptane

and alcohol of different strengths will, with the same notation as before, be as follows :—

Heptane :

$$2\,(1 + a)\,C_7\,H_{16} + 22\,O_2 + [82.8\,N_2] = A\,.\,CO_2 + B\,.\,CO + C\,.\,H_2O + D\,.\,H_2 + E\,.\,O_2 + [82.8\,N_2].$$

Alcohol :

$$2\,(1 + a)\,C_2H_6\,O + 6\,O_2 + [22.6\,N_2] = A\,.\,CO_2 + B\,.\,CO + C\,.\,H_2O + D\,.\,H_2 + E\,.\,O_2 + [22.6\,N_2].$$

The method of calculation is exactly the same as was explained for benzene, and need not be further particularised.

The results are as shown in Tables XI. and XII.

TABLE XII.—**Alcohol.**

Deg. of richness a	Composition of products of combustion.						Vol. before combustion.	Vol. after combustion.	Vol. ratio k.	Mean vol. heat per mol. mixt.	Heat of combustion $+1.59$.	Max. temp. ° C.
	N_2	CO_2	CO	H_2O	H_2	O_2						
0	22.6	3.03	6.97	5.85	0.15	0.56	30.6	33.16	1.084	7.71	18.56	2507
0.1	22.6	2.93	1.47	6.34	0.26	0.265	30.8	33.8	1.100	7.85	19.05	2524
0.2	22.6	2.67	2.13	6.76	0.44	0.085	31.0	34.7	1.120	7.92	19.05	2505
0.3	22.6	2.31	2.89	7.04	0.76	0.02	31.2	35.6	1.142	7.98	18.58	2429
0.4	22.6	1.98	3.62	7.21	1.19	0.00	31.4	36.6	1.165	8.00	17.98	2348
0.5	22.6	1.72	4.28	7.24	1.72	0.00	31.6	37.6	1.190	7.95	17.32	2278

VI.

COMPARISON OF RESULTS WITH THE DIFFERENT FUELS.

The calculated maximum explosion temperatures, allowing for dissociation, given in Tables X., XI., and XII. for different mixture strengths are collected together for purposes of comparison in Table XIII.

The figures given in Table XIII. are shown graphically in Fig. 5, in which explosion temperature is plotted against mixture strength.

It will be seen that with each fuel, when dissociation is taken into account, the maximum temperature is not reached when the mixture is "correct" ($a = 0$), as would be the case if there were no dissociation (see Fig. 4). The highest temperature occurs with mixtures

10–20 per cent richer than this. Moreover, the calculated explosion temperature varies but little over a wide range of mixture strength, especially in the case of benzene.

The explosion temperature may be expected to give a rough indication of relative power output with the different fuels. A more interesting basis of comparison is, however, obtained by multiplying temperatures by the volume or molecular number ratio (k in the

TABLE XIII.—**Explosion Temperatures.**

Degree of richness.	$a-0.5$	-0.20	0	0.1	0.2	0.3	0.4	0.5
Benzene .. .. °C.	1994	2470	2655	2690	2701	2681	2630	2561
Heptane .. .. °C.	—	—	2591	2623	2603	2527	2448	2366
Alcohol .. .. °C.	—	—	2507	2524	2505	2429	2348	2278

Tables) before and after explosion, since explosion pressures will be proportional to this ratio as well as to the explosion temperature. The value of k is considerably higher for heptane and alcohol than for benzene ; on the other hand, the thermal capacity of the products of combustion of one gram-molecule of original mixture is higher also. As regards power output, the two effects to some extent cancel

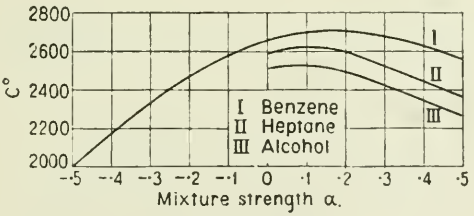


FIG. 5.
Explosion temperatures.

one another, the one lowering the explosion temperature, the other raising the pressures in relation to temperatures reached.

Table XIV. shows the product values of $k \times$ (rise of temperature on explosion) for conditions similar to those of Table XIII. The *rise* of temperature has been chosen instead of the actual explosion temperature, since this has been found to give values closely proportional to the net work done per cycle, *i.e.*, to (expansion work) − (compression work).

The figures of Table XIV. are plotted in Fig. 6 with mixture strength as abscissa.

It will be seen that, as between benzene and heptane (curves I. and II.), the lower explosion temperatures with the paraffin are

practically conterbalanced by the larger volume ratio; if the ordinates of Fig. 6 can be taken as a basis of comparison between the fuels, the maximum power output would not be expected to differ to any great extent. The ordinates $k\ (t_e - 372)$ of Fig. 6 vary even less over a wide range of mixture strength than do the

TABLE XIV.

Degree of richness.	$a-0.5$	-0.20	0	0.1	0.2	0.3	0.4	0.5
Benzene	1633	2150	2375	2440	2480	2497	2485	2451
Heptane	—	—	2408	2459	2480	2448	2406	2363
Alcohol	—	—	2315	2370	2390	2350	2303	2270

explosion temperatures of Fig. 5; the curves all show comparatively flat maxima, but of the three the power on benzene is noticeably better maintained with the very rich mixtures.

Now, these results all fall well into line with the most reliable engine test results:—

(i.) All tests agree in showing that the power of a petrol engine varies only slowly with the mixture strength. The power curve

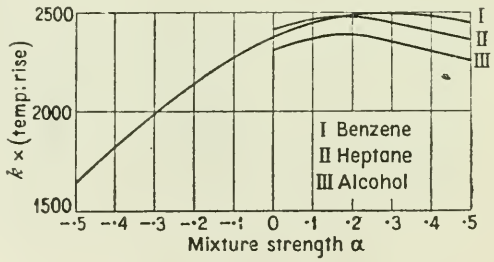


FIG. 6.

Product of volume-ratio and temperature rise on explosion.

shows a very flat maximum, like the curves of Fig. 6, not a particular optimum mixture as in the upper curve of Fig. 4.

(ii.) Maximum power is not obtained usually with the most economical mixture. The relation between economy and output depends to some extent on engine and carburettor, but tests examined show a rapid rise of power till the mixture is some 20 per cent richer than at greatest economy, and then a slow falling off as the proportion of fuel to air is increased beyond this point. By "maximum economy" is meant the point at which fuel per indicated horse-power per hour is a minimum.

(iii.) Reliable tests do not show any appreciable variation of maximum power as between aromatic and paraffin fuels.

The agreement of points (i.) and (ii.) above with the calculated results, based on fundamental measurements of calorific value, dissociation constants, etc., amounts to fairly conclusive evidence of the existence of dissociation. Further comparisons will be given later between calculated and observed thermal efficiencies, from which an estimate can be made of the quantitative accuracy of the calculations of dissociation, and explosion temperatures. The equality of maximum power to be expected from a paraffin and an aromatic fuel is a result which appears at first sight to be controverted by some published test results, but is confirmed by all the most reliable tests in the authors' experience.

VII.

Calculation of Final Temperature and Work Done During Expansion.

On the assumption that no dissociation takes place, this was given, for the correct mixture of benzene and air, in Section II. of Part II., p. 19. It was found that after an explosion temperature of 3023° C. adiabatic expansion in the ratio 5 : 1 gave a final temperature 2025° C., and that the work done on the expansion stroke was

8.94 kg. cal. per gram-mol. of original mixtures.

This corresponds to net work per cycle $= 8.94 - 1.59 = 7.35$, and thermal efficiency

$$\frac{7.35}{20.45} = 35.9 \text{ per cent.}$$

It is necessary now to see how these figures are affected by the existence of dissociation. It has been found in Part II., Section III., p. 26, that the explosion temperature of the dissociated mixture is 2655° C. There are two ways of considering the process of expansion. It may be assumed in the first place that no recombination of the dissociated mixture takes place during expansion, in which case, as we know the specific heats of all the gases concerned, we can calculate a new energy curve for the products of combustion (line 1, Table IX.), and also the final temperature and work done in exactly the same way as was done for the hypothetical undissociated mixture.

The result of this calculation gives the final temperature $T_f = 1709°$ C., and work done $= 7.94$ kg. cal.

The net work is therefore $7.94 - 1.59 = 6.35$, and thermal efficiency $= \frac{6.35}{20.45} = 31$ per cent, *i.e.*, about 15 per cent lower than the efficiency, if no dissociation occurred.

This figure should give a lower limit. The true efficiency (assuming no heat-loss) must lie somewhere between 31 per cent and 35.9 per cent.

If recombination takes place continuously during the expansion stroke, *i.e.*, if equilibrium is attained at each moment as the temperature falls, as seems highly probable, it is theoretically possible to calculate the work done from a knowledge of the equilibrium coditions for

$$p\,d\,v = \mathrm{C}\,d\,t + q_1\,d\,x,$$

where C is the *true* thermal capacity of the mixture and x the degree of dissociation at temperature T of the CO_2 (neglecting that of the H_2O, which is small). q_1 is the heat of dissociation of the CO_2.

x is given by the relation

$$\frac{x^3\,\mathrm{P}}{2\,(1-x)^2} = \mathrm{K}, \text{ where} \frac{d}{d\,t}\,(\log_e \mathrm{K}) = -\frac{q_1}{\mathrm{R}\,\mathrm{T}^2}.$$

The calculation is, however, too complicated to carry out rigorously, but an approximate result, to the degree of accuracy required, may be obtained in the following manner.

The effect of dissociation and recombination in the products of combustion is to make these behave as though their specific heat were, at high temperature, very much above its true value. The highest temperature reached is diminished and the fall of temperature during expansion is less rapid. The mixture of gases has, in fact, an "*apparent* specific heat" which will not differ from its true specific heat at low temperatures, but above about 2000° abs. will increase very much more rapidly.

We have, then, to imagine an energy-temperature curve for the products of combustion of the benzene mixture based on apparent specific heats corresponding to the actual dissociation conditions which obtain during expansion from 2655° C. This energy curve will be identical with that for the undissociated mixture at low temperatures (say up to 1500° abs.), but above 2000° abs. the energy will rise very rapidly with the temperature until it reaches 22.05 kg. cal. at 2655° C., instead of at 3023° C., as it would do if based on *true* specific heats (see Part II., Section II.). It would not be possible actually to draw this energy curve until the temperature at the end of expansion is known, since in calculating the dissociation conditions we have to begin to allow for diminishing volume at this temperature.

Probably the best method of getting at the final temperature, and so the work done on expansion, is as follows :—

A final temperature, T_f, is guessed, and the corresponding value of the dissociation, for which all necessary data exist, is calculated. The energy content of the mixture E_{T_f} can then be found, and if T_f is the true final temperature, we have

$$\text{Work done} = E_{T_e} - E_{T_f}.$$

Moreover, the mean *apparent* specific heat of the mixture during expansion is

$$\frac{E_{T_e} - E_{T_f}}{T_e - T_f}$$

Corresponding to this there is a definite value of the index n for the adiabatic expansion $P V^n = \text{constant}$, and the final temperature calculated from the relation

$$\frac{T_e}{T_f} = 5^{n-1}$$

must agree with the value of the final temperature originally guessed.

Actually, of course, the specific heat, and therefore n, is varying all down the expansion curve, and this method involves the assumption that the work done, or the area under the expansion curve, will not be sensibly different from that obtained on the assumption of a constant mean value of n. This point has been investigated in Appendix II., p. 44, for the undissociated mixture. An example is given in Appendix IV., p. 46, showing the possible error in the present case.

The only difference between the calculation of the equilibrium condition at the end of the expansion stroke and that given for explosion in Part II., Section III., p. 26, is that in this case the volume is that before compression.

The partial pressure of oxygen therefore $= \frac{E}{73.4} \times \frac{T_f}{373}$, or, if x and y are the numbers of molecules of CO_2 and H_2O dissociated so that the products of combustion are $(12 - x)\,CO_2 + x\,CO + (6 - y)\,H_2O + y\,H_2 + \frac{x+y}{2}\,O_2 + 56.4\,N_2$, the partial pressure of oxygen is

$$\frac{x+y}{2} \cdot \frac{1}{73.4} \cdot \frac{T_f}{373} = Po_2,$$

whence

$$\frac{x^2}{(12-x)^2} \cdot \frac{x+y}{2} \cdot \frac{1}{73.4} \cdot \frac{T_f}{373} = K_1;$$

also we have

$$\frac{x}{(12-x)} \frac{(6-y)}{y} = K,$$

so that, since K and K_1 are known in terms of the temperature, corresponding values of x, y, and T_f can be found in the neighbourhood of the final temperature. This will probably fall near 2000° C., and taking a series of values round about the point we get Table XV.

TABLE XV.

T_f° abs.	x.	y.	$\frac{q_1 x + q_2 y}{73.4}$
2177	0.60	0.039	0.59
2242	0.80	0.053	0.78
2299	1.00	0.067	0.98
2247	0.817	0.054	0.87

In the last column, q_1 and q_2 are the heats of dissociation of CO_2 and H_2O, so that the figures tabulated represent the potential chemical energy of the mixture, due to its dissociated condition. They give, in fact, the amount by which the energy curve of the mixture corresponding to *apparent* specific heats lies above that calculated from *true* specific heats.

Columns 2, 3, and 4 can be plotted against the temperature and then by trial the temperature, T_f, is found which fulfils the necessary conditions set out above, namely, that

$$\frac{T_e}{T_f} = 5^{n-1},$$

where n is found from the mean apparent specific heat of the mixture between T_e and T_f.

It has been shown in Part II., Section II., that the energy curve based on true specific heat is very closely represented, between 2000° C. and 3000° C. by the equation

$$E_T = 0.549 \left(\frac{T - 373}{100}\right) + 0.00703 \left(\frac{T - 373}{100}\right)^2.$$

Hence that based on apparent specific heats will be

$$E_T{}^1 = 0.549 \left(\frac{T - 373}{100}\right) + 0.00703 \left(\frac{T - 373}{100}\right)^2 + \left[\frac{q_1 x + q_2 y}{73.4}\right],$$

x and y having the appropriate values for the temperature T.

The following calculations illustrate the method; they are given for the correct final temperature, found by trial to be 2247° C. absolute.

The values of x, y, and $\frac{q_1 x + q_2 y}{73.4}$ corresponding to $T = 2247°$ C. are obtained from plotted curves and are given at the foot of Table XV.

The energy in the gas at the end of expansion is

$$E_{2247} = 0.549 \times 18.74 + 0.00703 \times (18.74)^2 + 0.800 = 13.55.$$

$\therefore$ work done on expansion $= E_{2928} - E_{2247} = 22.05 - 13.55 = 8.50$ kg. cal.

$$\text{Mean specific heat} = \frac{8.50}{681} = 12.48 \text{ cal. per degree.}$$

whence

$$n = \frac{12.48 + 2.05^*}{12.48} = 1.1642$$

* No. of mols. at beginning of expansion = 1.04 per mol. of original mixture (col. 9, Table X., p. 28).

No. of mols. at end of expansion $= \frac{74.83}{73.4} = 1.02$ per mol. of original mixture.

Mean value of "R" $= 1.985 \times 1.03 = 2.05$ as taken above.

and $$T_f = \frac{T_e}{5^{0.161}} = \frac{2928}{1.303} = 2247^\circ = 1974^\circ \text{ C.}$$

The calculation made on the assumption that no recombination of the dissociated mixture took place during expansion, of which the results were given at the beginning of this section, showed a temperature fall during expansion from 2655° C. to 1709° C. and work done equal to 7.94 kg. cal.

Allowing for recombination has therefore increased the calculated work done from 7.94 to 8.50, and the final temperature from 1709° to 1974° C. In the mixture at the end of expansion there is still a certain amount of CO_2 and H_2O in a dissociated condition, namely :

$$\frac{0.817}{12} \times 100 = 6.8 \text{ per cent of the total } CO_2$$

$$\frac{0.054}{6} \times 100 = 0.9 \text{ per cent of the total } H_2O.$$

The thermal efficiency is

$$\frac{8.50 - 1.59}{20.45} = 33.8 \text{ per cent,}$$

as against 31 per cent if no recombination had taken place, and 35.9 per cent if there had been no dissociation at all.

The results for the three cases may be tabulated as in Table XVI. Compression ratio 5 : 1.

TABLE XVI.

	Explosion temperature.	Final temperature.	Net work.	Efficiency.
	° C.	° C.		
Assuming no dissociation ..	3023	2013	7.35	35.9
Dissociation but no recombination during expansion ..	2655	1709	6.35	31.0
Dissociation and recombination	2655	1974	6.91	33.8

We have, then, a thermal efficiency of 33.8 per cent for the standard engine of comparison working at compression-ratio 5 : 1, when the actual conditions of the working substance are allowed for. Such an engine, in fact, should reproduce actual engine conditions except as regards heat-loss to the cylinder and irregularities of combustion. The best absolutely reliable figure for indicated thermal efficiency at 5 : 1 compression which the authors have been able to obtain is rather under 32 per cent, or 70 per cent of the air-standard efficiency. This leaves, when compared with 33.8 per

cent, practically no margin for the heat-loss. But it has already been pointed out that observed thermal efficiencies are calculated when the engine is running on approximately a 20 per cent weak mixture.

In Table X., p. 28, the explosion temperature for such a mixture was given as 2470° C., allowing for dissociation. A further calculation similar to that just made for the correct mixture gives a final temperature after expansion of 1707° C., and net work per cycle 6.24 kg. cal. per gram-mol. For a 20 per cent weak mixture the whole heat of combustion is 16.46 kg. cal. per gram-mol., and the efficiency of the standard engine with this weak mixture is therefore

$$\frac{6.24}{16.46} = 0.379.$$

The margin left for heat-loss to the cylinder during explosion and expansion is at least 14 per cent, on the assumption that the experimental efficiency, 0.32, was obtained on a 20 per cent weak mixture (see Appendix V., p. 47). Since a large part of the heat-loss to the cooling water will take place during the exhaust stroke and round the exhaust-valve seating when the gases have left the cylinder, this figure of 14 per cent during expansion, when compared with an average total jacket loss of 25 per cent, is of the right order. Although more exact data as to mixture strength at the point of maximum economy would be desirable, it can be said that the calculated thermal efficiencies are reasonable and afford a satisfactory check on the estimated conditions in the cylinder.

It is possible, with a suitably designed combustion head, to burn, as a stratified charge, a mixture which would be too weak as a homogeneous mixture. Such a process has great advantages as a method of running at less than full load owing to the gain of thermal efficiency by low-temperature working. Conditions were therefore calculated for a 50 per cent weak mixture, and are shown in Table XVII.

TABLE XVII.—**Explosion Temperatures, Thermal Efficiencies, etc., at 5 : 1 Compression, allowing for Dissociation and Recombination.**

Mixture strength.	Explosion temp., ° C.	Final temp., ° C.	Net indicated work.	Thermal efficiency.
Correct	2655	1974	6.91	0.338
20 per cent weak	2470	1707	6.24	0.379
50 per cent weak ..	1994	1153	4.27	0.412

As will appear from Table X., about 20 per cent of the CO_2 is dissociated at the beginning of expansion in the 20 per cent weak mixture. By the end of expansion, however, the dissociation has fallen to 0.4 per cent of the CO_2, and the heat thrown away in the exhaust gases is, in consequence, practically the same as if no dissociation

had taken place. On the other hand, if a 50 per cent weak mixture could be exploded, dissociation would be negligible even at the maximum temperature reached.

VIII.

The Effect of an Increase in Compression-ratio.

The opinion has been expressed by some engineers that the "air-standard" efficiency is a misleading guide as regards the effect on thermal efficiency of an increase in compression-ratio; that although at normal ratios practical thermal efficiencies are a constant fraction of air-standard values (for the same sized engines), yet at high compression-ratios there will not be an increase proportionate to that of the air-standard and that a definite useful limit will soon be reached.

As regards *brake* thermal efficiencies, there will, no doubt, be a practical limit to the useful compression-ratio owing to the necessary increase in the weight of moving parts and consequent increase in mechanical losses. It will be interesting, however, to calculate the effect on maximum temperature and calculated thermal efficiency of an increase in compression-ratio, and to see whether there is any reason to expect, as has been suggested, that the indicated thermal efficiency will reach a limit on purely thermodynamic grounds.

The following calculations have been made for the same mixture of benzene and air as that of line 3, Table X., p. 28, but it has been supposed compressed in the ratio of 10 : 1 before explosion instead of 5 : 1.

Under these conditions the energy added by adiabatic compression is 2.59 kg. cal. per mol. of mixture, and the temperature at the end of compression is 544° C.

The composition of the products of combustion and the explosion temperature are given in Table XVIII. side by side with the corresponding figures for the same mixture compressed in the ratio 5 : 1.

Table XVIII.—**Compression-ratio 10 : 1.**

Deg. of richness	Composition of products of combustion.						Vol. before combustion.	Vol. after combustion.	Vol. ratio k.	Mean vol. heat per mol. mixt.	Total energy added.	Explosion temp. °C.
	N_2	CO_2	CO	H_2O	H_2	O_2						
5.0	56.4	8.22	3.78	5.80	.20	1.99	73.4	76.4	1.040	7.20	18.39	2655
10.0	56.4	8.17	3.83	5.77	.226	2.03	73.4	76.4	1.040	7.27	19.31	2756

In spite of the increased pressure the dissociation is slightly increased, with the result that the rise of temperature on explosion

is less than with 5 : 1 compression. It remains now to examine how the work done and thermal efficiency are affected.

The dissociation conditions are worked out for a few temperatures in the neighbourhood of the probable final temperature, and the true temperature found, as before, by trial. Since the explosion temperature is not very much higher, the final temperature will, for this large expansion-ratio, be considerably lower than for the 5 : 1 expansion. It is found to be 2023° C. abs., or 1750° C. as against 1974° C. for the 5 : 1 expansion.

At the end of the 10 : 1 expansion 2.4 per cent of the CO_2 still remains dissociated, and about 0.35 per cent of the H_2O. The total "energy" in the gases at the end of expansion is 11.27 kg. cal. per mol. of original mixture.

The work done is therefore

$$2.59 + 20.45 - 11\ 27 = 11.77$$

and the thermal efficiency

$$\frac{11.77 - 2.59}{20.45} = 44.9 \text{ per cent.}$$

The effect of increasing the compression-ratio is brought out clearly in Table XIX., in which the thermal efficiencies, after allowing for dissociation, etc., are compared with the "air-standard" efficiencies.

TABLE XIX.

r	Air-standard efficiency, per cent.	Estimated thermal efficiency if no heat-loss.	Relative efficiency to air-standard, per cent.
5	47.5	33.8	71.2
10	60.2	44.9	74.6

This comparison brings out the very interesting fact that the indicated thermal efficiency may be expected to approach more nearly to the air-standard efficiency the higher the compression-ratio; or, in other words, that on thermodynamic and chemical grounds the advantage to be gained by using a high compression-ratio is actually greater than the air-standard efficiencies would suggest. Mechanical difficulties may limit the utility of increasing compression-ratios, but apart from this, the above result seems to dispose of the view that there is any limit to the thermodynamic advantage of high compression-ratios. All calculations have, of course, been made on the assumption of no heat-loss, but it is not likely that this will be seriously increased when it is considered that no more than about 100° C. is added to the explosion temperature by such a large increase of compression as from 5 : 1 to 10 : 1.

IX.

Conclusions.

It has been shown that it is possible, starting from the available data as to true specific heats and high temperature equilibrium conditions, to calculate the temperature at each point of the cycle of an internal combustion engine when the composition of the fuel and the mixture strength are known, and no loss of heat occurs.

It has also been shown that, in spite of the higher calorific value of paraffin petrols, the maximum powers to be expected from a paraffin and an aromatic fuel are not appreciably different when compared at a compression-ratio which allows full ignition advance without detonation on both fuels. On the other hand, it is well known that higher compression-ratios can more safely be used with aromatic than with paraffin fuels, and it has been shown in the last Section that on thermodynamic grounds it is advantageous to push the compression-ratio up as high as possible. With a suitably designed engine, therefore, the aromatic fuel will hold a decided advantage. The curves of Fig. 6, p. 32, show that as regards maximum power output alcohol should be definitely inferior to the other two fuels, but this result postulates the same initial temperature of charge. In practice, the initial temperature will be considerably lower with alcohol than with other fuels under similar working conditions, since the latent heat is much higher.

The form of the curves in Fig. 6 brings out clearly the effect of dissociation in depressing the maximum power available at the correct mixture and giving fairly constant power over a wide range of mixture strength. But the ordinates of these curves are proportional to the power calculated on the assumption that no recombination takes place during expansion; they considerably over-estimate the effect of dissociation. In the light of Part III., Section VII., p. 33, it is possible to make a more accurate estimate of this at all mixture strengths.

The curves for benzene are reproduced in Fig. 7. It will be seen that at the correct mixture the depression of the maximum power is about 14 per cent as between curves I. and III.; the actual figures for the net work done per cycle are 7.35 and 6.35 kg. cal. per mol. of mixture (Table XVI., p. 37).

It was shown in Part III., Section VII., that when recombination is allowed for the net work is 6.91, represented by the point B on Fig. 7, just 6 per cent below the maximum point A, for no dissociation.

At all mixture strengths, curve I. will define the upper and curve III. the lower limit of the power available, and the intermediate dotted curve drawn to the right from B can be given with certainty

as a close approximation to the actual power available with rich mixtures when recombination is allowed for. On the weak side, curve I. will still be the upper limit of power, to which the curve allowing for dissociation will rapidly approach. As pointed out in Section VII., by the time the mixture is 20 per cent weak there is no appreciable difference between the two curves. A number of points on this curve could, of course, be calculated, as the point B has been, but the position of the curve is so well defined by the position of curves I. and III. that the labour of calculation would be superfluous.

Taking, then, the dotted curve as being the nearest estimate of work per cycle, it will be seen that when allowance is made for recombination the result is, on the rich side, an even greater con-

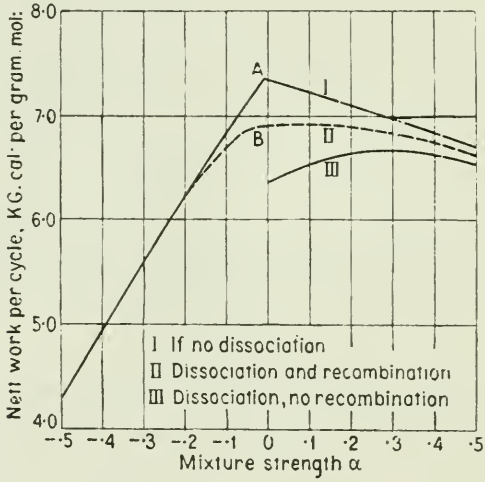


FIG. 7.
Power output. Benzene.

stancy of power with mixture strength; there is no appreciable variation up to $a = 0.25$. On the weak-mixture side, however, since curve I. must still give the upper limits of power, there will be a very rapid falling off. This is exactly what is found in engine tests: as the mixture is weakened below the point of maximum power there is at first a very slight falling off, but a point is soon reached at which the fall of power becomes rapid and running unstable, owing to the irregularity of burning.

On Fig. 8 the theoretical curves have been replotted, and also the results of a test carried out by Mr. H. R. Ricardo and the authors.

The compression-ratio in this test was 5 : 1, and the fuel was benzene of 98 per cent purity. The relative position of the curves II.

and III. is indefinite, as the quantity of air taken in was not measured. The *shapes* of the curves are, however, comparable (*i.e.*, the percentage power variation for the same fractional change

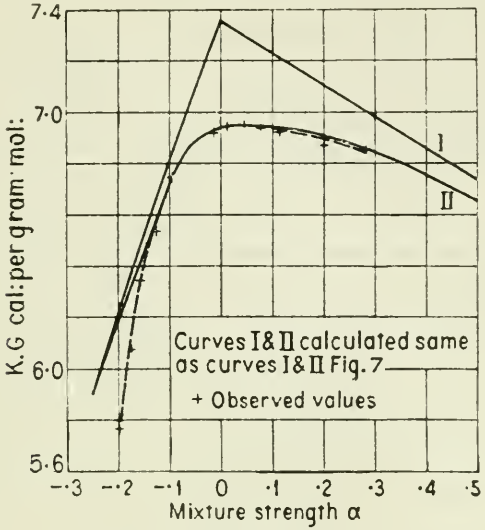


FIG. 8.
Power output. Benzene.

of mixture strength) and the similarity is sufficiently remarkable to afford substantial confirmation of the calculated results.

APPENDIX I.

The volumetric heat of the products of combustion of a " correct " mixture of benzene and air is, for the temperature range during compression, 5.83 cal. per gram-mol. of original mixture. This corresponds to a value of n, in the adiabatic compression curve $P V^n = k$, of 1.3405.*

The absolute temperature, T_B. at the end of compression will be

$$375 \times 5^{0.3405} = 645.$$

The Centigrade temperature is 372°, and the energy added on compression

$$272 \times 5.83 \times 10^{-3} = 1.585 \text{ kg. cal. per gram-mol.}$$

* The volume of the gram-mol. is taken as 22.41 litres (Nernst). This corresponds to a value of the gas constant R = 1.985, whence

$$n = 1 - \frac{R}{C_v} = 1 - \frac{1.985}{5.83} = 1.3405.$$

The mean volumetric heat of air for the same temperature range is very nearly 5.17 cal. per gram-mol., and that of benzene vapour about 30.

The volumetric heat of the mixture is

$$\frac{5.17 \times 71.4 + 30 \times 2}{73.4} = 5.84 \text{ cal. per gram-mol.,}$$

which corresponds to $n = 1.3400$.

The compression temperature is lowered 0.5 degree C., and the energy added is not appreciably changed.

Appendix II.

On the error introduced by assuming a mean value for γ during the expansion.

The correct equation for T_f is

$$R \ln 5 = 4.965 \ln \frac{T_e}{T_f} + 0.001406 (T_e - T_f)$$

where $$R = 2.012.$$

or, multiplying by 0.4343,

$$R \log. 5 = 4.965 \log. \frac{T_e}{T_f} + 0.0006106 (T_e - T_f)$$

$$R \log. 5 = 2.012 \times 0.699 = 1.406.$$

If $T_f = 2290$, the right-hand side of the equation is

$$= 4.965 \times \log. 1.439 + 0.0006106 \times 1006$$
$$= 0.7845 + 0.6142$$
$$= 1.399.$$

If $T_f = 2280$, $\frac{T_e}{T_f} = 1.446$; $\log. 1.446 = 1.602$.

Right-hand side $= 4.965 \times 1.602 + 0.0006106 \times 1016$
$= 1.416.$

$\therefore$ True value for $T_f = 2286 = 2013°$ C.,

as against 2017° C. obtained by using mean value for γ. Work done $= 8.98$ *kg. calories*, as against 8.94, *i.e.*, 0.45 per cent more than that calculated by assuming a mean value for γ.

Appendix III.

Details of Maximum Temperature Calculation.

In the reaction

$$2\ C_6H_6 + 15\ O_2 + 56.4\ N_2$$
$$= A.\ CO_2 + B.\ CO + C.\ H_2O + D.\ H_2$$
$$+ E.\ O_2 + 56.4\ N_2$$

we have

$$\begin{aligned} A + B &= 12 \\ 2A + B + C + 2E &= 30 \\ C + D &= 6. \end{aligned}$$

And, if the maximum temperature be guessed as 2650° C.,

$$\frac{B^2}{A^2} E = 0.421 \text{ and } \frac{B \cdot C}{A \cdot D} = K = 13.4.$$

The best way of solving these five equations is probably to guess a value for one of the unknowns, say E, deduce A, B, C, and D, and see if the reaction constant K = 13.4 is satisfied.

A little practice will show to what extent E must be modified to fulfil this condition. K varies comparatively slowly with the temperature, and it will be found that an error in the reaction constant corresponding to 300 degrees C. difference of temperature makes a negligible difference (about 0.05 per cent) in the heat liberated on combustion and in the final result.

Hence it is not necessary to go on adjusting A, B, C, and D until K corresponds very exactly with the final temperature calculated.

The solution for A, B, C, D, and E gives

$$\begin{aligned} A &= 8.22 \\ B &= 3.78 \\ C &= 5.80 \\ D &= 0.20 \\ E &= 1.99. \end{aligned}$$

If the whole two molecules of benzene had been burnt to CO_2 and water, the heat of combustion of the mixture would have been 2×751 kg. cal. or $\frac{2 \times 751}{73.4} = 20.46$ kg. cal. per gram-mol. of mixture.

But since there are in the final mixture 3.78 molecules of CO and 0.20 molecule of H_2, the actual heat liberated will be (2×751) less the heat of combustion of these quantities of CO and H_2.

The heat liberated, therefore, per gram-mol. of mixture will be

$$\frac{2 \times 751 - 3.78 \times 68 - 0.20 \times 57.8}{73.4} = 16.80 \text{ kg. cal.}$$

and the energy content at maximum temperature = 16.80 + 1.59 = 18.39 kg. cal.

There are, in the products of combustion of 73.4 gram-mol. of original mixture,

Diatomic gases 56.4 + 3.78 + .20 + 1.99	= 62.37
CO_2	= 8.22
H_2O	= 5.80

The mean volumetric heats 100° C. – 2750° C. (from Table I., p. 4) are:

Diatomic gases	6.09	cal.	per	gram-mol.	per	°C.
CO_2	10.89	,,	,,	,,	,,	,,
H_2O	10.16	,,	,,	,,	,,	,,

Hence the mean volumetric heat of the products of combustion of 1 gram-mol. of original mixture between 100° C. and 2650° C. is

$$(62.37 \times 6.09 + 8.22 \times 10.90 + 5.8 \times 10.16) \div 73.4 = 7.20 \text{ cal. per °C.},$$

and the calculated maximum temperature is

$$\frac{18390}{7.20} + 100 = 2655° \text{ C.}$$

Appendix IV.

The Accuracy in Taking a Mean Value of n in $P V^n$ Const. for Calculating the Work Done During Expansion when Recombination is Allowed for.

It was shown in Appendix II. that for the expansion of the undissociated mixture it was quite sufficiently accurate to take a mean value of n corresponding to no net gain or loss of heat during expansion.

In order to show that this is also justifiable when recombination is taking place, it has been thought advisable to deal with the expansion in stages, calculating the equilibrium condition at three points during the process, and the work done during each stage.

Starting with the dissociated mixture at 2655° C., the equilibrium condition, and hence the energy content (heat and chemical), of the mixture has been found after expansion to twice, three times, and five times the original volume.

The results are shown in Table XX.

Table XX.

Vol.	Temp. = ° C.	Energy content.	Mean specific heat for range.	Work done.
1	2655	22.04		
			12.99	3.935
2	2352	18.105		
			12.68	2.105
3	2186	16.00		
			11.55	2.46
5	1973	13.54		

It will be seen that the final temperature reached, after calculation in three stages with the correct specific heat for each range, only

differs by 1 degree from that found when a mean specific heat of 12.48 cal. per gram-mol. per degree was taken for the whole range, and that the calculated work done is identical with that found in Section VII., namely, 8.50 kg. cal.

Appendix V.

In Part I., Section IV., p. 11, it was stated that the maximum theoretical efficiency, assuming no loss of heat during the working stroke, was 38 per cent at a compression-ratio of 5 : 1, whereas the best experimental figure actually obtained was 32 per cent. A small part of this difference must doubtless be attributed to the fact that the explosion is not instantaneous, but the major part is probably due to loss of heat during the working stroke. Some estimate of this from the difference between the theoretical and practical result can be obtained in the following way. The total heat in the working fluid (reckoned from 100° C.) at the maximum pressure = heat of combustion + heat of compression = 16.46 + 1.59 = 18.05 kg. calories. The work done on expansion is 6.24 + 1.59 = 7.83 kg. calories, and the efficiency of expansion, reckoned on total heat, is therefore $\frac{7.83}{18.05}$, or about 43 per cent. The efficiency of expansion reckoned in this way increases somewhat with decrease of temperature, but this can be neglected. If the entire heat-loss occurs before the expansion begins, and if it is equal to a fraction x of the heat of combustion, the total heat = $18.05 - Q\,x$, where Q = heat of combustion, and the work done on expansion is then

$$0.43\,(18.05 - Q\,x).$$

Hence the loss in efficiency

$$= 0.43 \times Q\,x = 0.38 - 0.32$$

or

$$x = \frac{0.06}{0.43}\,Q$$

$$= 0.14\,Q.$$

Hence the difference between the maximum theoretical and practical efficiency is accounted for by a loss of heat equal to 14 per cent of the total heat of combustion before expansion begins. The observed loss of heat to the jackets in these experiments was 25 per cent of the heat of combustion. This is the loss during the complete cycle, so that the difference between theory and experiment is satisfactorily accounted for, especially when it is remembered that the further the expansion goes before the loss of heat occurs, the smaller will be the effect of this on the work done.

SECTION II.

RICARDO'S INVESTIGATIONS FOR THE ASIATIC PETROLEUM CO.

THE INFLUENCE OF VARIOUS FUELS ON THE PERFORMANCE OF INTERNAL COMBUSTION ENGINES.

In August, 1919, the directors of the Asiatic Petroleum Co., Ltd., invited the author to carry out, on their behalf, an extensive series of experiments on volatile hydrocarbon fuels. After some discussion it was decided to extend the experiments to embrace as thorough an investigation as possible into the behaviour of the working fluid in an internal combustion engine cylinder, and more particularly the influence of the composition of the fuel upon its behaviour.

The Asiatic Petroleum Co. all along desired that the line of research should be such as would yield results of real scientific interest, and made it clear from the start that they had no wish to restrict the range of research within limits which would be of direct commercial value to themselves alone. Further, they readily agreed that, in order to render the results really conclusive, every endeavour should be made to ensure the highest attainable standard of accuracy throughout, and to this end they were prepared to provide the most costly and elaborate apparatus.

The author was fortunate in being able to obtain, in addition to the staff of his own firm, the assistance of Mr. H. T. Tizard and Mr. Marshall, of Oxford University, and Mr. D. R. Pye, of Cambridge University, all of whom have contributed invaluable assistance, not only in investigating the chemical properties of the various samples tested and making valuable recommendations, but also in analysing the experimental results obtained, and in carrying out a very exhaustive theoretical examination of the properties of the working fluid under all conditions.* The author is also deeply indebted to Mr. Kewley, chief chemist of the Asiatic Petroleum Co., Ltd., for his wholehearted assistance and enthusiasm. His great knowledge of the chemical problems involved and his generous co-operation throughout proved invaluable. Great credit is due also to Mr. O. Thornycroft, who throughout had direct charge of all the experimental apparatus, and to whose knowledge and care the accuracy and consistency of the experimental results are largely due.

At the date of this report (February, 1921) a very large amount of experimental work had been carried out, and many data had been obtained both from practical tests and from theoretical reasoning.

* See p. 1 *ante*.

Although much yet remains to be done and many avenues are still awaiting exploration, it was considered that sufficient information had been acquired, and a sufficient number of conclusions had been arrived at, to warrant the publication of, at any rate, an interim report.

Before proceeding further, the author would like to pay a tribute to the very public-spirited attitude of the Asiatic Petroleum Co. in permitting him to publish the results so far obtained. The research has necessarily involved the design and manufacture of much intricate and highly specialised apparatus, and the continuous employment of a very considerable staff of experts. It will be realised, therefore, that it has been of a very costly nature.

1.—Analysis of Existing Information.

Before any actual experimental work was started upon a careful analysis was made of the existing information with a view to indicating the direction in which research was likely to prove most fruitful, and at the same time a preliminary investigation was made both as to the requirements which an ideal fuel should fulfil and as to the possible variations which might be expected as between fuels of different chemical composition.

The analysis of existing information proved, for the most part, somewhat disappointing; although many valuable data relating to the theoretical side were found, particularly in the publications of the *Journal of the Chemical Society*, and in various other scientific publications, the actual experimental data available proved to be of very little value. In many of the published experimental results it was found, on close scrutiny, that the results would not agree among themselves; such data had to be set aside as unreliable. In other cases, although careful cross-checking revealed the fact that the experiments were carefully and intelligently carried out, it revealed also, in nearly every case, that the performance of the engine on which the investigations were made was so low, either in power or efficiency, or both, that some large factor was clearly intervening to prevent the attainment of anything approaching the efficiency of which the engine should be capable. This factor was generally bad distribution or bad combustion-chamber design, and, since the influence of either of these factors, and more particularly the first, will vary widely with fuels of different volatility or chemical composition, it was obvious that the controlling factor was a variable one as between different fuels, and therefore must vitiate the results. The outcome of this analysis was therefore to reveal that, so far as engine test results were concerned, there appeared to be no information sufficiently accurate or reliable to be of much practical value.

2.—Lines of Investigation.

After much thought it was decided that the most fruitful lines of investigation to explore, as a beginning, were :—

(1) To make a thorough investigation into the factors controlling detonation, since the tendency to detonate determines the degree of compression which may be employed, and therefore both the power and efficiency of an engine.

(2) To determine accurately the total internal energy of each fuel when mixed with the quantity of air required for complete combustion, for when burnt with equal efficiency it is clear that the power output obtainable with each fuel will be directly proportional to the total internal energy. It is perhaps as well to emphasise that the internal energy of the cylinder charge has little or no connection with the heat value of the fuel, in terms of its weight.

(3) To determine the losses due to change in specific heat and dissociation at high temperatures, and to ascertain how far these losses vary with different fuels.

(4) To investigate the available range of burning of different fuels.

(5) To investigate the factors controlling distribution and ease of starting.

(6) To determine accurately the calorific value of different fuels.

(7) To determine accurately both the self-ignition temperature and the process of combustion, when different fuels are ignited by adiabatic compression in a large cylinder.

Of these factors, it was already realised that the first was likely to prove by far the most important—it was known in a very general way that certain fuels, such as benzole and other members of the aromatic group, also alcohol, would not detonate readily even under very high compression-pressures, and that detonation could be kept in check by the addition of benzole to ordinary petrol—a fair amount of data was available also as to the phenomena of detonation. Very little information of a quantitative nature was, however, available, nor was anything known as to the influence of other fuels upon detonation.

With regard to (2), (3), and (4), very little information was available, at all events in a form readily applicable by engineers. Sir Dugald Clerk, Professor Hopkinson, and other members of the Gaseous Explosions Committee of the British Association, had, before the war, made a very extensive investigation of these factors in relation to coal gas, but it has long been realised by many investigators that the conditions are widely different when applied to the volatile hydrocarbon fuels which contain no inert constituents. It is, of course, evident that any variation either in the total internal energy, in the losses due to dissociation and change of specific heat,

or in the range of burning available, would influence very powerfully both the power output and efficiency obtainable, even with the same compression-ratio.

With regard to (5), it was clear that the distribution and ease of starting must necessarily be dependent to some extent at least on the chemical and physical characteristics of the fuel, and it was clearly of importance to determine how far it was so dependent, and to ascertain what were the practical limits as to range of boiling point, vapour tension, and other such considerations.

With regard to (6), the published data as to the heat value of different fuels were found in many cases to be contradictory and unreliable.

Finally, as regards (7), a somewhat elaborate piece of apparatus has been designed and constructed for research in this direction, but few data are as yet available.

3.—General Conclusions from Tests.

Before proceeding to details either of the apparatus employed or of the experimental or theoretical results obtained, it will perhaps be well to outline very briefly the definite practical conclusions which have already been arrived at, and to deal later and in more detail with the reasoning and the experimental results by which they have been deduced and confirmed.

With the exception of alcohol and certain other bodies, these conclusions may be taken as applying to all volatile liquid hydrocarbons practically available and all combinations thereof. Alcohol and other bodies containing oxygen atoms in the molecule differ in certain important characteristics, and therefore cannot be included without some reservations. The general conclusions, expressed as briefly as possible, subject, as such conclusions always must be, to minor qualifying conditions which will be dealt with later, are as follows :—

(1) It has been proved that the tendency of a fuel to detonate is the one outstanding factor in determining its value for use in an internal combustion engine. Compared with this, all other considerations are of secondary importance.

(2) There appears to be little doubt as to the correctness of the view, now generally accepted, that detonation is less the lower the rate of burning of the fuel.

(3) In all cases it seems that a low rate of burning is advantageous. No fuel has yet been found of which the rate of burning was too low to permit of maximum efficiency being obtained in the highest-speed engine yet tested.

(4) Fuels capable of standing a very high compression will operate in a low-compression engine equally as efficiently as those of which

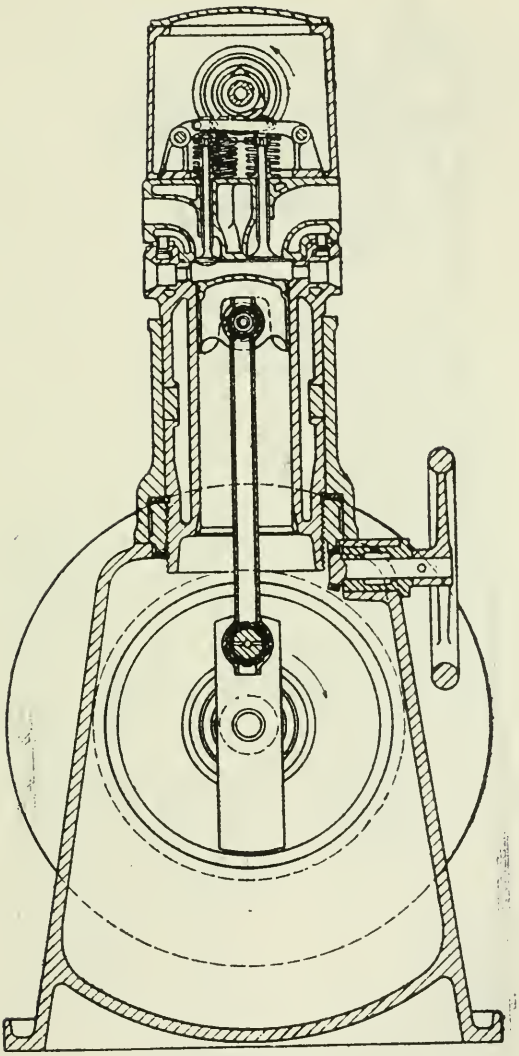


FIG. 1.
General arrangement of variable-compression engine.

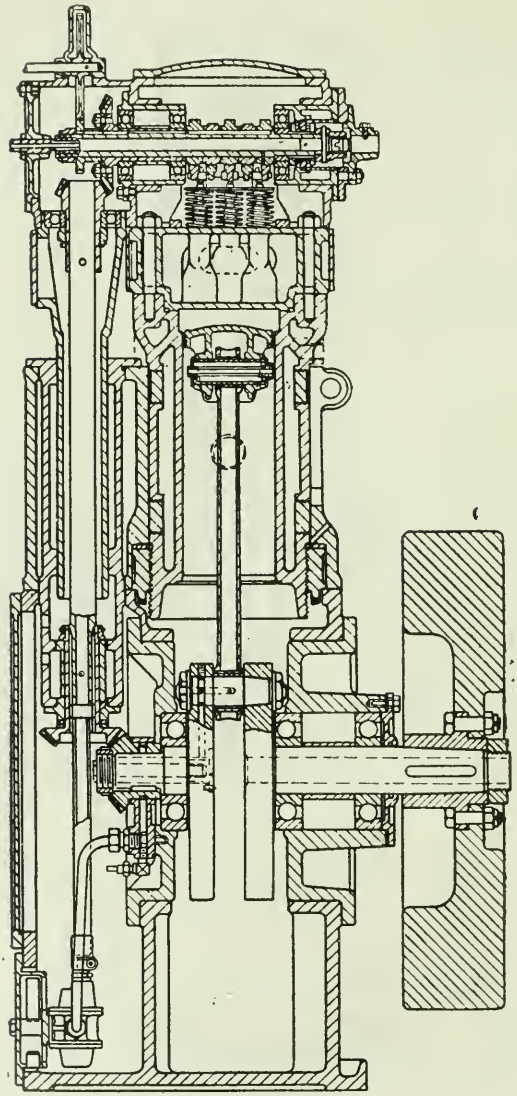


Fig. 2.
General arrangement of variable-compression engine.

the normal rate of burning is high, provided that there is a reasonable degree of turbulence in the combustion chamber.

(5) Apart from the limitations introduced by detonation, the power output obtainable from all volatile liquid fuels, with the exception of the alcohol group, is the same at the same compression to within less than 2 per cent. Such variations as occur within this range are due rather to variations in the latent heat of evaporation than to any other circumstance.

(6) Owing to the high latent heat and low boiling point of alcohol and certain other bodies, the weight of charge per cycle is greater and a higher power output is obtained in consequence.

(7) The efficiency with which all volatile fuels, other than alcohol and the others previously referred to, are burnt is the same, at the same compression-ratio, irrespective of rate of burning, provided that the compression is low enought to avoid detonation under any circumstances. In the case of alcohol the efficiency is very slightly higher, for reasons which will be considered later.

(8) The useful range of burning is to all intents and purposes the same for all volatile liquid fuels.

(9) The unavoidable losses due to dissociation and change of specific heat at high temperatures are substantially the same in all cases.

(10) All the experimental results indicate that, with certain trifling exceptions which will be considered later, the performance of any combination of hydrocarbons as regards detonation, and therefore the power output and efficiency obtainable, is the mean performance of each of the components. The performance of any complex fuel such as petrol can therefore be predicted, once the nature and proportion of its constituents are determined, or conversely, a fuel can be prepared to give any required performance within the limits available.

(11) No increase in power or efficiency has been found or need be expected from the use of "dopes." A large number of these were investigated and tested, but, as might be expected, the improvements in all cases proved to be practically negligible, but certain substances, notably tetra-ethyl lead, when added in small quantities, tend to suppress detonation.

(12) The highest useful compression-ratio for, and therefore the power output obtainable from, any petrol is governed by the relative proportions of aromatics, naphthenes, and paraffins it contains—the smaller the proportion of the latter the better from almost every point of view. The influence of olefines has not yet been investigated.

(13) To judge of the quality of a fuel by its specific gravity is entirely misleading. If naphthene and aromatic fractions are present in any large proportion (as is sometimes the case), then a high specific gravity is a substantial advantage.

(14) Owing to the very rich mixture delivered normally by pilot jets and the still further enrichment effected by flooding, only a

relatively small proportion of highly volatile constituents appears to be required for starting.

(15) There is considerable evidence to indicate that in the case of most "natural" petrols the more valuable constituents are those of which the boiling points lie between the range 80° C. to 180° C. It is therefore often quite as undesirable to restrict the final boiling point of a fuel to 150° C. as it is to permit it to exceed 200° C. So much, however, depends on the actual composition of the fuel, and so great are the variations in this respect that this conclusion must be regarded only as a broad generalisation.

4.—List of Apparatus.

The apparatus constructed or requisitioned for these experiments will be described in detail at a later stage; the principal items consist of:—

(1) A special variable-compression research engine, Figs. 1 and 2. This engine has been described in a paper presented by the author at the British Association meeting at Cardiff and reported in full in *Engineering* of September 3rd and 10th, 1920.

(2) A research engine provided with means for supercharging the cylinder.

(3) A special machine for determining the self-ignition temperature and pressure of fuels under adiabatic compression.

(4) A single-cylinder experimental engine with crosshead piston, used chiefly for experiments on fuels of high boiling point.

(5) A more or less floating population of engines of various commercial types, mostly four-cylinder, used for tests on starting, check tests, etc.

(6) Several motor cars of different types for road testing.

(7) A miscellaneous collection of auxiliary apparatus.

It will be shown later, in detail, that the standard of accuracy obtainable with both the research engines, and more particularly the variable-compression engine, is exceptionally high; extreme variations extending over a year's running have been found to be less than 1 per cent, while the mean results obtained during any one group of tests may be taken as accurate in most respects to within one-quarter of 1 per cent.

The absolute accuracy of all the figures given later as test results may safely be accepted as being within one-half of 1 per cent, while the relative accuracy is about twice as high.

The accuracy of tests on the various commercial engines is not of a very high order. While the accuracy of observation is comparable with that obtained on the research engines, the mechanical and thermal efficiency of such engines is, for the most part, so low that variations in the efficiency or power output obtainable with different

fuels were only too frequently completely obscured by much larger variations in the friction losses, distribution, carburation, etc.

Little was hoped for from road tests on cars, and indeed they proved to be practically valueless for indicating small differences. It was found impossible to assess and allow for the variations due to road surface, wind, tyre inflation, and many other variables, all of so large a magnitude as completely to mask any but well-marked differences.

It soon became apparent that the only conclusions of any value from road tests would be those drawn from the mean results of a large accumulation of data obtained from the behaviour of many vehicles operating over a long period on each sample of fuel. Such a test was not practically possible.

5.—Characteristics of Fuels Tested.

One of the most serious difficulties which had to be contended with was that in many cases it was altogether impracticable to obtain chemically pure samples in sufficient bulk. It would, of course, have been desirable to use a pure light fraction —preferably of the paraffin series—as a standard basis of comparison, but to obtain a pure fraction, such as hexane, in the quantity required for calibration and check testing, would have been altogether out of the question. After much consideration it was decided to employ as a standard basis of comparison a comparatively light petrol containing a fairly high proportion of paraffin fractions, and from which all, or almost all, of the aromatic members of the series had been removed by sulphonation. For this purpose the Asiatic Petroleum Co. set aside some 5 tons of suitable petrol and treated it by sulphonation until the aromatic content was reduced to less than 2 per cent. This fuel, which was used throughout the whole series of experiments, both for calibrating the engines and for comparative purposes, will be referred to in future as standard aromatic free petrol. Its general characteristics are given in Tables I. and II.,* pp. 142–145.

From these Tables it will be seen that the standard aromatic free petrol consists almost entirely of members of the paraffin and naphthene series. Since the whole quantity was drawn from one tank and sulphonated at one operation, it follows that its uniformity does not vary—a very important consideration in view of the fact that the research was expected to occupy not less than two years, and any change in the composition of the standard fuel would have involved endless complications and the possibility of serious errors.

It was found that the minimum quantity of fuel required for any individual test was about two gallons, and this, of course, left no margin for re-tests or for checking discrepancies, such as are inevitable when working within such fine limits. While every effort was

* Note.—As so many references to the same Table are made throughout this Section of the Report, it has been thought well to place all the Tables together at the end, on pp. 142 to 154.

made to obtain chemically pure samples of the various members of different series, it was by no means always possible to do this, and in several instances serious errors were introduced due to the presence of impurities in what purported to be chemically pure samples. Of the paraffin series, reasonably pure hexane was, with great difficulty, obtained in sufficient bulk to permit of extensive tests. Of the naphthene series, samples of cyclohexane, hexahydrotoluene, and hexahydroxylene were obtained in sufficient bulk, but, although the first of these purported to be chemically pure, subsequent analysis indicated that it contained so considerable a percentage of paraffins and aromatics as appreciably to modify the conclusions drawn from the earlier experimental results. Of the aromatic series, samples of benzene and toluene of great purity were relatively easily obtained. The very greatest difficulty was experienced in obtaining ethyl alcohol, and, in spite of numerous applications, the Customs authorities resolutely declined to permit of its being supplied duty free, in spite of the fact that the nature and purpose of the research was perfectly well known to them.

Considerations of the chemical and physical properties of ethyl alcohol made it clear, not only that it would prove highly interesting in itself, but also that its behaviour might be expected to throw a good deal of light on several questions relating to other fuels, such as the very important one of the influence of the latent heat of evaporation. Fairly early in the course of the research the necessity for running tests on alcohol became evident, and means had therefore to be found for obtaining sufficient quantities of this most interesting fuel. The samples ultimately obtained proved to be of a very high degree of purity.

Of the olefine series no really pure samples have yet been obtained in quantities sufficient to run engine tests.

Of the sulphur compounds, both carbon bisulphide and methyl mercaptan were obtained in sufficient quantities, but the influence of thiophine has not yet been investigated.

With the other miscellaneous samples, such as ether, acetylene, acetone, etc., no difficulty was encountered in obtaining sufficient supplies of reasonable purity.

For the most part the impurity of the samples has not had any very important influence on the deductions to be drawn from the test results, but their presence has rendered the tests far more arduous, since it was necessary in each instance to determine the nature and proportion of the impurities present and to arrive by independent tests at their influence. Once these factors had been determined the influence of their presence on the behaviour of the fuel as a whole could be allowed for.

After the preliminary ground has been covered both the theoretical and experimental results, and the conclusions to be drawn from them, will be considered in detail.

The close agreement between the theoretical deductions arrived at by Messrs. Tizard and Pye* and the actual experimental data has been throughout a very striking feature. The influence of sulphur compounds of low ignition temperature upon the tendency to detonate, and the somewhat eccentric behaviour of heptane, have been the real surprises, while the very important influence of the latent heat of evaporation upon the power output obtainable was not appreciated at first, nor does it appear to have been realised by any other investigators.

6.—Series of Tests Undertaken.

The actual series of tests may be divided broadly into two sections :—

Section 1.—A series of comparative tests carried out on a large number of different samples of petrol with an immediate view to determining the best mixtures for blending, and indirectly with a view to exploring and determining the most fruitful avenues for further research. This series was carried out during the earlier stages of the research, while the more accurate and intricate apparatus was in the process of design and construction, and while the theoretical investigations were in progress.

Section 2.—A much more accurate and analytical series of tests carried out with apparatus designed specifically for the purpose, using pure, or relatively pure, samples of all the different ingredients of the various volatile liquid fuels, conducted with a view to

(1) Checking and proving the accuracy of the theoretical determinations.

(2) Ascertaining the individual characteristics of samples of each of the various leading groups and the influence of the presence of such samples on the performance of a composite fuel.

(3) Determining accurately the influence of compression-ratio, temperatures, and pressure, upon the power output and efficiency of an engine.

(4) Determining the influence of the addition of gases, both active, such as acetylene, and inert, such as nitrogen, CO_2, etc.

In view of the large number of readings taken it is impossible to give actual individual results, except in the cases of certain more or less isolated experiments. For the most part, the curves and Tables which will be given represent the mean results of many hundreds of readings.

In Table I., p. 142, are given a list and the leading characteristics of the various samples obtained in sufficient bulk to permit of fairly extensive tests. In addition to those given in this list, a large number of other samples of petrol and also a number of relatively pure substances were tested, but these have been omitted here—in

* *Ante.*

the former case because the results obtained from them were not of sufficient interest, and in the latter because the quantity obtainable was not sufficient with which to run a full range of tests.

The various samples of petrols are selected either because they are fairly representative or, in some cases, because they afford extreme instances either as regards their specific gravity, aromatic content, naphthene content, etc. For obvious reasons it is not desirable to give the trade names of these different petrols, but they include samples of well-known commercial petrols of all grades, ranging from the lightest aviation spirit to the heaviest grade used for mechanical transport.

The degree of purity of the so-called pure substances is denoted by the figures in brackets.

Tests on 95 Per Cent Alcohol.

In Section III. of this volume, and following the description of the research outlined above, is a series of tests carried out at the request of the Empire Motor Fuels Committee. The primary object of this research was accurately to determine the behaviour of alcohol and alcohol-benzole mixtures in the internal combustion engine already mentioned, which is equipped essentially for research purposes and provided with a means for varying the compression-ratio while running.

Prior to the carrying out of this research, the engine was fitted with a new combustion head giving a somewhat higher combustion efficiency than that used previously. For this reason both the power output and thermal efficiency are under all conditions higher than the figures which are shown in the research carried out for the Asiatic Petroleum Co., Ltd.

The fuel supplied for the tests was ethyl aclohol, 95 per cent by volume, and having a specific gravity of 0.815 at 15° C.

Series of Tests Undertaken.*

As the result of meetings with Dr. Ormandy and Mr. G. W. Watson representing the Empire Motor Fuels Committee, it was agreed that the following series of tests should be carried out :—

Series I.—Tests for power output and consumption over the complete available range of mixture strength with wide-open throttle at piston speeds of 800, 1200, 1600 and 2000ft. per minute and compression-ratios of 3.8 : 1 and 7.0 : 1. Constant heating to carburettor. Number of tests 8. Similar check test with compression-ratio of 5.0 : 1 and piston speed of 1600ft. per minute. Number of tests 1.

Series II.—Throttle tests for power and efficiency over complete range of mixture strength at 0.8, 0.6 and 0.4 full load

* For tests on alcohol-ether mixtures see p. 243.

at piston speeds of 1200 and 2000ft. per minute and with a compression-ratio of 5.0 : 1. Constant heating to carburettor. Number of tests 6.

Series III.—Tests over complete range of mixture strength at a piston speed of 2000ft. per minute. Compression-ratio 5.0 : 1. Constant heating to carburettor. Jacket outlet temperatures 30° C. and 90° C. Number of tests 2.

Series IV.—Tests over complete range of mixture strength at a piston speed of 2000ft. per minute. Compression-ratio 5.0 : 1. Heat input to carburettor nil and 2000 watts. Number of tests 2.

Total number of tests 19.

It was further recommended that in the case of Series II. a directly comparative series of tests should be run on petrol.

GENERAL DESCRIPTION OF ENGINE AND TESTING GEAR.

The Variable-compression Engine.

The variable-compression engine used for the investigations is illustrated in Figs. 1 and 2. It is essentially a high-duty engine capable of running at very high speeds and yields an exceptionally high power output and thermal efficiency. Very particular care has been taken to eliminate all variable factors so far as it is possible to do so, and to ensure the highest possible standard of consistency. The general lay-out of the testing plant is shown diagrammatically in Fig. 3, while Fig. 4, Plate I., is a photograph taken from the dynamometer end.

Means were provided for varying the compression of the engine over any range from 3.7 : 1 up to 8 : 1 while running at full power, and without disturbing any temperature, frictional, mechanical, or other conditions.

The combustion chamber is so designed that its general form and ratio of surface to volume undergo the minimum of alteration when the compression is varied.

Special means are adopted to render the engine as little sensitive as possible to changes in the temperature of the lubricant. Ball bearings are used wherever possible, in order to reduce variation in friction with different oil temperatures, and the water jacketing round the barrel of the cylinder is stagnant, and therefore quickly attains a constant temperature, independent of the temperature of the supply. This ensures that the piston friction which is dependent upon the temperature of the oil on the cylinder walls reaches a minimum in the course of a few minutes, and thereafter remains constant. The importance of retaining the same general form of combustion chamber under all conditions of compression cannot be over-estimated. Very misleading results have been obtained when

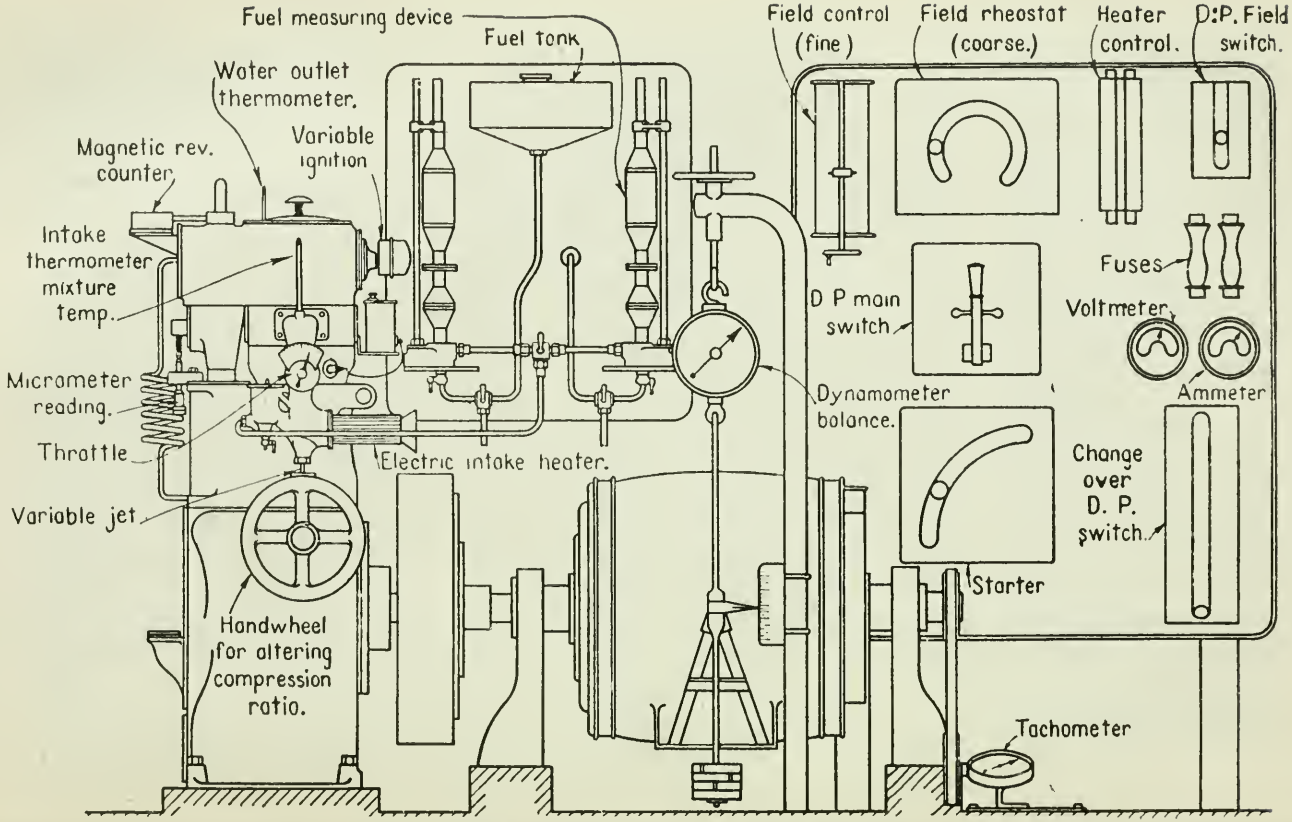


FIG. 3.—General arrangement of testing apparatus.

the compression-ratio has been varied by fitting different pistons, some with concave, others with convex crowns. In one series of experiments with different compression-ratios which the author examined, and which were obviously carried out with the most scrupulous care, the results were entirely vitiated because the whole character and efficiency of the combustion chamber was completely changed, as between the low compression and the high, with the result that a certain optimum compression-ratio was claimed to have been found, after which any further increase in compression resulted in loss of power and efficiency. A careful scrutiny of the results showed that at or near the so-called optimum compression-ratio the efficiency of the combustion chamber was at a maximum, and that at the higher compressions it fell away very rapidly, and indeed became quite exceptionally inefficient.

In the variable-compression engine designed for the purpose of these tests the efficiency of the combustion chamber undergoes very little change between the lowest compression-ratio and the highest, with the result that the efficiency increases with increase of compression at a perfectly normal rate throughout the whole range.

As will be seen from the sectional drawings Figs. 1 and 2, the compression-ratio is varied by raising or lowering the whole cylinder, together with the carburettor, camshaft, and valve gear; by this means the compression-ratio can be varied over any range in the course of a few seconds, and without disturbing any of the temperature conditions or any adjustments.

To measure and record the compression-ratio in use a micrometer is provided, as shown in Fig. 5, Plate II. This is arranged to operate electrical contacts, and controls a pilot lamp which lights up immediately the desired compression is reached, so that the operator can adjust the micrometer screw at his leisure to the compression-ratio he requires before making any alteration, and can then see at a glance by the lighting up of the lamp that this compression has been reached.

For the ignition of the charge four sparking-plugs are fitted equidistant round the circumference of the combustion chamber, each of which is connected to a Remy high-tension coil. The low-tension circuit of all the coils is operated by a single Remy contact-breaker driven directly from one end of the camshaft. The object of using this arrangement in preference to magnetos was twofold :—

(*a*) To ensure that the passage of the spark across all four plugs should be absolutely synchronous.

(*b*) To ensure that the intensity of the spark should be the same at all settings.

In practice it was found that the use of two sparking-plugs on opposite sides of the combustion space gave equally good results, and all tests were therefore run under these conditions.

Measurement of Load and Speed.

The power is absorbed by means of a balanced swinging-field electric dynamometer, the torque being recorded by means of an open-scale spring balance reading in one-tenths of a pound. A dead weight, slightly in excess of the maximum torque of the engine, is suspended from a torque arm and the spring balance therefore records only the difference between the dead weight and the actual torque of the engine. The load on the engine, and therefore the speed also, are controlled by varying the field excitation by means of two rheostats connected in series, one giving a relatively coarse adjustment, and the other a very fine one. The speed is determined approximately by means of a large dial tachometer which is accurate to within plus or minus 1 per cent; for all fuel-consumption and other readings upon which an accurate determination of the speed depends, a magnetically-operated revolution counter is used. This counter is driven from the camshaft through the medium of a magnetically-controlled jaw-clutch, and by this means the exact number of revolutions during any given time or during the consumption of any given quantity of fuel can at all times be counted and recorded.

Carburation and Consumption.

The Claudel-Hobson carburettor is provided with a fine-adjustment needle-valve in the main jet, by means of which the mixture strength can be controlled to any desired degree. The air entering the carburettor can be heated by means of a series of electrical heater elements placed in a well-lagged passage close behind the jet. The quantity of heat supplied to these elements is recorded by means of a voltmeter and ammeter mounted on the switchboard, and can be adjusted by means of a continuous coil rheostat. By this method it is possible to ensure and maintain a constant heat-input ranging up to 2000 watts or 114 B.Th.Us. per minute. For the purpose of these tests, and in order to eliminate any variables, a standard heat-input of 0.0433 B.Th.Us. per revolution per minute was used (except when expressly stated to the contrary), corresponding, at the normal speed of 1500 revs. per minute, to 65 B.Th.Us. per minute.

For the measurements of fuel consumption a special fuel-measuring device is used (Fig. 6, Plate III.), consisting of a vertical double conical chamber, tapering to a very narrow and parallel neck at either end and provided with a gauge glass. Markers on the gauge glass on a level with the restricted portion of the neck indicate a capacity of 142.1 c.c., or exactly one-quarter of a pint. The fuel falls rapidly through the narrow necks at the commencement and completion of the measured quantity, and its passage past the markers can be clocked with great accuracy; at the same time the magnetic counter

is thrown into operation as the fuel passes the first marker, and is thrown out, and a brake applied automatically to its spindle, as it passes the second marker. Thus both the time taken to consume a given quantity of fuel and the actual number of revolutions made during the consumption are always recorded.

Procedure for Tests.

When making tests for fuel consumption under any given set of conditions the procedure adopted is as follows :—

After the engine has been run up to speed and all conditions have become normal, a few check readings are first taken on one or other of our standard fuels, depending upon the compression used. So soon as the readings obtained with the standard fuel are found to be within 0.5 per cent of the normal value for that fuel under the particular conditions of the test the supply is changed over to the fuel under test, the carburettor adjusted to yield a fairly strong mixture and the best ignition setting found. After any conditions influenced by the nature of the fuel, such as the water temperature etc., have been re-adjusted, the following readings are taken :—

(1) Revs. per minute by tachometer (observed but not recorded).
(2) Torque.
(3) Heat-input to carburettor.
(4) Temperature of outside air.
(5) Water outlet temperature.
(6) Gas temperature in induction pipe.
(7) Ignition setting as recorded on a dial attached to the contact-breaker.
(8) Time in seconds during consumption of measured quantity of fuel.
(9) Exact number of revolutions made during consumption of measured quantity of fuel (by automatic counter).

After taking one such set of readings the supply of fuel to the jet is reduced by slightly closing the needle valve, and, if necessary, the ignition timing and the field resistance of the dynamometer are readjusted to suit the altered conditions brought about by the use of a slightly weaker mixture. This done, another and similar set of readings is taken, after which the mixture is again weakened. This procedure is continued until the mixture strength becomes too weak for steady running, with the result that the speed fluctuates slightly.

As a general rule about ten such sets of readings are taken at every test, ranging from the mixture giving maximum power down to the weakest that can be used.

At the conclusion of each test or group of tests the fuel and circulating water supplies are simultaneously cut off, and the engine is motored round at the same speed, the change over from full power

to motoring being effected almost instantaneously without either stopping the engine or appreciably reducing its speed. A reading is then taken of the torque required to rotate the engine at the speed at which the test has been run, and under precisely the same temperature conditions.

The object of this latter test is to ascertain whether the internal friction of the engine has undergone any change.

Since the value of any test depends, not only upon the degree of accuracy with which it is conducted, but still more upon a definite estimation of this degree, it is well to state clearly the direction and magnitude of any errors which may be introduced.

Accuracy of Readings.

The general degree of accuracy of the readings shown on the test sheets and curves may be taken to be as follows :—

(*a*) *Torque.*—The average total weight lifted by the torque arm is about 36 lb. It can be read with certainty to within $\pm$ 0.1 lb. on the spring balance, and when the voltage on the field circuit is steady, to within $\pm$ 0.05 lb. As regards torque, or, as it is expressed, "mean pressure," the accuracy may therefore be regarded as being always within the limits $\pm$ 0.3 per cent and generally within $\pm$ 0.2 per cent.

(*b*) *Heat-input to Carburettor.*—This is read in terms of volts supplied to the heater. Both the voltmeter and the resistance of the heater coils have been checked against standard instruments, and their readings, when corrected for known errors, may be taken as accurate to within $\pm$ 1.0 per cent.

(*c*) *and* (*d*) These are direct thermometer readings and are probably accurate to within about 1 degree C. in absolute terms and $\pm$ 0.25 degree C. relatively.

(*e*) The readings of temperature of the gases flowing through the induction pipe between the carburettor and the inlet valves are taken by means of a thermometer lying in an insulated pocket filled with mercury fitted into the induction pipe. Owing to the fact that the walls of this pocket are bombarded with finely divided liquid particles of fuel, there is good reason to believe that this thermometer behaves to a large extent as a wet-bulb instrument, and that the absolute values of its readings are therefore unreliable. Readings taken at this point must therefore be regarded as being of relative value only. In view of the fact that under no conditions was the whole of the fuel evaporated the relative values are probably fairly reliable.

(*f*) *Ignition Timing.*—This is recorded on a dial by means of a long pointer attached to the contact-breaker. The absolute relationship between the divisions on the dial and the crank-angle (which varies slightly as the platinum points wear down)

is checked at the conclusion of every test. The accuracy therefore of the ignition timing (by which is meant the breaking of contact between the platinum points of the contact-breaker and not the actual time of passage of the spark) may be taken as ± 0.5 degree of the crank-angle.

(*g*) *Time in Seconds and Calibration of the Measuring Device.*—Experiments with different observers and as between different observations by an individual observer indicate that the rapid fall of fuel in the gauge glass past the starting and finishing marks can be observed to within 0.2 second by any individual observer, and to within 0.4 second as between several observers. The average time when running at 1500 revs. per minute on full throttle with a reasonably economical mixture may be taken as 40 seconds. The accuracy of observation so far as timing is concerned may therefore be taken as ± 0.25 per cent during any one series of tests, or ± 0.5 per cent as between different tests, when different observers have been employed. So far as the tests for the Empire Motor Fuels Committee are concerned, all the readings of fuel consumption were taken by the same observer throughout. As to calibration, the measuring device has been re-calibrated from time to time by different observers, all of whom have found its capacity to be the same to within ± 0.1 per cent.

(*h*) The magnetically-operated counter is geared to one-fourth engine speed; its maximum possible error is therefore 3 revolutions at the commencement and 3 revolutions at the conclusion of each test, a total of 6 revolutions in an average of 1200, or ± 0.25 per cent.

The Supercharging Engine.

The supercharging engine used in these experiments has been fully described in a paper read before the Institution of Automobile Engineers,* a brief description of its operation will therefore suffice.

For the purpose of these tests it was employed almost solely for investigating the tendency of fuels to detonate, and in order to make the detonation as severe as possible a somewhat higher compression-ratio was used than formerly, and at the same time the exhaust valve was arranged to open earlier in order to reduce to the minimum the proportion of cooled exhaust gas in the scavenge and supercharge air. From the point of view of investigating detonation it was found to be more sensitive than the variable-compression engine, also very accurate readings could be taken in a remarkably short space of time—an important consideration in all experimental work, more especially when dealing with samples of which the quan-

* See Proc. I.A.E., Vol. XV., p. 717.

tity was very limited, as was the case with many of the "pure" samples.

This engine has a cylinder of 4⅜in. bore and 5½in. stroke, and was intended to run normally at a speed of 1500 revs. per minute, at which speed all tests for detonation were carried out. The valves are arranged with the inlet directly over the exhaust, both valves being in a side pocket. It is fitted with a crosshead-type piston, and for the purposes of these experiments a fixed compression-ratio of 5.18 : 1 was used.

Supercharging is effected in the following manner:—Advantage is taken of the differential area of the piston and crosshead to pump air, which is drawn into a chamber below the piston through a series of very light bronze flap-valves; the air thus entrapped is compressed by the piston to a pressure of about 10 lb. per sq. in., and then delivered through a water cooled inter-cooler to the cylinder. The lower end of the cylinder barrel is perforated with a number of small ports just uncovered by the piston at the bottom of its stroke. The cycle of operations beginning with the induction stroke is as follows:—

As the piston descends a mixture of fuel and air enters the cylinder through the inlet valve in the usual manner, and at the same time the air in the chamber below the piston is compressed. At about 95 per cent of the downward stroke the ports referred to previously are uncovered by the piston and the air in the crosshead chamber enters the cylinder, after passing on its way through the water cooled inter-cooler, which removes from it a large proportion of the heat of compression and also of the heat picked up from contact with the underside of the piston. About the same time the main inlet valve to the cylinder is closed and the pressure in the cylinder rises, due to the addition of the supercharge air from slightly below atmospheric pressure to 5 lb. per sq. in. above. The additional air entering in this manner does not mix freely with the fuel and air in the cylinder, but forms more or less a layer over the crown of the piston. On the following upward stroke the contents of the cylinder are compressed, and at the same time a fresh supply of air is drawn into the crosshead chamber below the piston. During compression very little mixture between the live charge and the supercharge takes place. On ignition the inflammation of the first portion of the charge causes a local and rapid rise of pressure, and so sets up a fresh disturbance which causes complete or nearly complete admixture of the cylinder contents, but not until the first portion is fully ignited. During the expansion stroke the second charge of air drawn into the crosshead chamber is compressed. Towards the end of this stroke the exhaust valve opens and the bulk of the products of combustion are discharged. Shortly after the opening of the exhaust valve the ports in the cylinder wall are again uncovered by the piston, a second charge of cool compressed air then enters the cylinder,

driving before it some of the remaining products of combustion. During the following upward stroke of the piston the contents of the cylinder are discharged through the exhaust valve, leaving only the compression space full of a mixture of exhaust products and air. At the same time a fresh charge of air is drawn into the crosshead chamber in readiness for the supercharge.

From the above brief description it will be seen that the cylinder is alternately scavenged and supercharged by means of the air compressed below the working piston. The additional air provided in this manner per cycle is probably about 40 per cent. The supercharge air can be measured with a fair degree of accuracy and amounts to from 34 to 35 per cent of the total cylinder volume; the amount added due to the scavenging cannot so readily be assessed, but is probably in the neighbourhood of 5 per cent.

For controlling or cutting off the supercharge a movable shutter is provided, surrounding the ports in the cylinder wall in such a manner that they can be partially or completely closed. In the latter event the engine runs in every respect as a normal four-stroke engine, and the air in the crosshead chamber is alternately compressed and expanded. The shutter is so arranged that the supercharge can be introduced or cut off instantaneously. In so far as the testing gear is concerned the arrangements are almost precisely similar to those described previously in relation to the variable-compression engine, except that the carburettor heating is effected by means of a hand-controlled exhaust jacket and there is no magnetically-operated revolution recorder. Since, however, the supercharging engine has been employed almost exclusively for investigating the problem relating to detonation, these refinements have not been found necessary. In an engine running on this cycle with an excess of air always present in the cylinder the mean effective pressure varies directly as the quantity of fuel admitted up to the point at which the whole of the supercharge air is saturated; moreover, maximum efficiency is maintained over a very wide range of mean pressure. In this respect, therefore, the supercharging engine showed a very marked advantage over the variable-compression unit in which the mixture strength for maximum efficiency at each compression-ratio had to be searched for in every instance. By increasing the quantity of fuel admitted the mean effective pressure could be raised in direct proportion until a point was suddenly reached at which violent detonation was set up. In the case of every fuel tested which came within the range of this engine the mean pressure at which detonation suddenly occurred was not only very sharply defined (to within less than 1 lb. per sq. in.), but it was also found to be quite extraordinarily consistent.

This engine was constructed some six years ago for a totally different purpose, but it has proved invaluable for fuel research, and in spite of its age and the very rough usage to which continued

detonation experiments must subject any engine, it stood up to its work remarkably well for a period of eighteen months. It has not quite so high a standard of mechanical consistency as the variable-compression engine, and is rather more sensitive to variations in oil temperature, etc.; but, none the less, its performance has varied by less than 1.5 per cent at any time over a period of eighteen months' hard usage.

Miscellaneous Engines.

Apart from the special research engines, samples of many of the fuels were tested in other engines on the test beds. These consisted of two or three experimental engines of various types, permanently fitted up on test beds for general research work, and a more or less floating population of representative engines by various makers, undergoing investigation of one sort or another. All these engines, whether permanent or temporary, were direct-coupled to swinging-field electric dynamometers and fitted with means for recording both the power and economy with a very high degree of accuracy. From a general scientific standpoint the results obtained from a large number of engines of widely different types and sizes proved extraordinarily interesting; but in so far as the fuel problem was concerned they yielded little or no information which could not be obtained more readily and more accurately on the special research engines. Each engine, of course, had its own particular characteristics depending upon the details of its design—in some the distribution was good, in others bad, some had efficient combustion chambers, others the reverse, some detonated even with a very low compression, while others would stand a high compression without trace of detonation on the same fuel. Some had insufficient heating to the carburettors, others excessive. As a result of these individual characteristics each engine behaved differently, but all of them behaved perfectly rationally, and, once their idiosyncrasies were recognised and allowed for, the results obtained with each sample of fuel could be predicted to within very close limits from the experiments on the research engines. No single case occurred where the behaviour of a fuel differed as between that in one of the research engines or any other engine, when the characteristics of the latter were taken into proper consideration.

The experiments, however, on miscellaneous engines were not without value, for they indicated that the results obtained in the special research engines could be relied upon as being applicable to any other type, provided the varying conditions were taken into account, while they afforded generally a useful check. The experiments on standard engines, however, would have been of little or no value by themselves unless run in parallel with those on the special research engines. Without the latter they would have proved frankly

misleading in many instances. For example, before any accurate deductions could be drawn as to the behaviour of any particular fuel, it was found absolutely essential to determine definitely and accurately the rate of heat-input to the carburettor, for upon this depends not only the degree of volatilisation, but also the volumetric efficiency, and therefore the power output, and also to a considerable extent the tendency of the fuel to detonate. In all ordinary engines the carburettor and induction system receive heat either from the water circulation or from the exhaust pipe. In neither case is it possible either to determine the amount of heat so received or even to keep the amount constant, for readings of temperature either of the gas in the induction pipe or in the carburettor proved utterly unreliable, and were of value only when the rate of heat-input also was definitely known. It has been shown previously that the rate of heat-input to the carburettor, as opposed to the temperature of the gases, proved to be, in fact, one of the most important of all the controlling factors in the behaviour of a fuel.

The variation in the tendency to detonate as between different forms of combustion chamber was especially marked. In the case of "A" petrol, for example, this detonated in the variable-compression engine at a compression-ratio of 6 : 1. In the case of several of the other engines it detonated at ratios below 5 : 1, and in one extreme instance of bad combustion-chamber design it actually detonated with a compression-ratio of only 4.3 : 1. This particular engine could not run on any ordinary petrol on full load without the use of an over-rich mixture, and or a very late ignition timing, so that its fuel consumption on most grades of petrol was abnormally high, and it could only be run at a reasonable economy on petrols "A" and "H."

Detonation.

Previous experience had indicated very clearly that, not only was the tendency of fuels to detonate probably by far the most important of the factors controlling the power output and efficiency of an internal combustion engine of the explosion type, but also, that the tendency to detonate varied widely as between different fuels, and was likely to prove much the most important factor in determining their suitability or otherwise. This has proved to be the case. It was, of course, common knowledge that certain brands of petrol detonated more readily than others, that commercial benzol would not detonate under any circumstances in an engine of normal design, and that the addition of benzol to petrol always reduced its tendency to detonate. Beyond this little was definitely known. It was decided, therefore, to make as thorough an investigation as possible into the whole phenomenon of detonation.

1.—Influence of Detonation on Efficiency.

Before dealing with the influence of the nature of the fuel upon detonation, it will be well to consider the influence of detonation itself upon the performance of an engine.

All engines using volatile liquid fuels operate on a cycle in which a combustible mixture of fuel and air is drawn into the cylinder, is compressed, ignited, and then expanded until it occupies the same volume as before compression. From any given weight of mixture the power output and efficiency obtainable depend upon the degree of expansion, and since, in this cycle, the ratios of compression and expansion are equal, it follows that both power and efficiency are dependent upon the ratio of compression. Setting aside for the time being such questions as dissociation at high temperatures, change of specific heat and loss of heat to the cylinder walls, all of which reduce the absolute efficiency obtainable, but which do not alter materially the relationship between compression-ratio and efficiency, we find that the limiting theoretical efficiency is as given by the formula $E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$, where r is the compression-ratio and γ is the ratio between the specific heat of air at constant pressure and at constant volume.

Fig. 7 shows the relationship between efficiency and compression over the range from 3.5 : 1 up to 7.5 : 1 according to this formula, which is known as the Air Standard Efficiency, and is the accepted basis of comparison.

It will be observed that the air standard efficiency rises rapidly at first as the compression-ratio is increased, but the rate of gain in efficiency diminishes considerably with the higher compression-ratios. It is doubtful whether, but for exceptional cases, any material advantage would be obtained from the use of a compression-ratio higher than 7 : 1, on account of the relatively small gain in efficiency obtainable thereby, and the very high maximum pressures involved. It would necessitate a considerable increase in the cost and weight of the engine to enable it to withstand these high pressures, and an increase, also, in the internal friction due to the heavier reciprocating and rotating parts.

The compression-ratio which can actually be employed is governed in practice by the tendency of the working fluid to detonate under the high pressure and temperature of compression.

2.—The Nature of Detonation.

The phenomenon of detonation appears to be substantially as follows. When a mixture of hydrocarbon vapour and air is compressed to a high pressure and to a temperature approaching that of its self-ignition temperature, and is then ignited from any one

point, the flame at first spreads by the normal process of combustion, compressing before it the unburnt portion of the charge. When the rise in temperature of the unburnt portion due to compression by the burning gases exceeds the rate of heat dissipation by a certain margin, spontaneous ignition of the former takes place, and an explosion wave is set up which strikes the walls of the cylinder with a hammer-like blow, thus causing the familiar metallic ring known as "pinking" or "detonation." It also has the effect of compressing and still further raising the temperature of the portion first ignited, and with it the temperature of the sparking-plugs or other partially insulated objects in their immediate vicinity. If detonation be allowed to persist it will, unless it be very slight, increase in severity and ultimately raise the temperature of the plug points to such a degree as to cause pre-ignition, or self-ignition of the

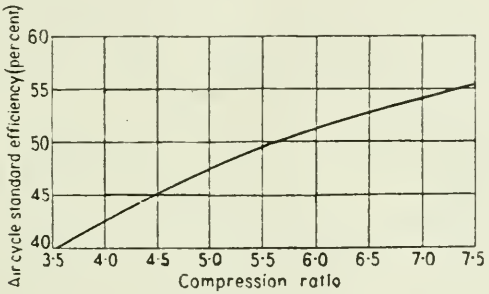


FIG. 7.—Relation between the air standard efficiency and compression-ratio : (total volume ÷ clearance volume).

charge, during compression and before the spark has passed; this is quite a distinct phenomenon and must not be confused with detonation.

It will be seen that detonation depends upon the normal rate of burning of the fuel, and occurs only when the rate of spread of flame is appreciably greater than the rate at which the unburnt portion can get rid of its heat. It is dependent also to a lesser degree upon a number of other factors which will be considered shortly.

Before proceeding further, it is well to point out that there are several stages of detonation, depending upon how far the normal spread of combustion has extended through the mass of the working fluid before spontaneous ignition occurs. Detonation, in its incipient stages, is sometimes very difficult to detect, because it is often unaccompanied by any audible noise, but rather by what may be termed harsh running.

3.—Tests with an Increasing Compression-ratio.

The following very interesting and instructive experiment was frequently carried out.

A fuel was selected which, in the variable-compression engine, was known to detonate with a compression-ratio of, say, 5 : 1. The compression was set at, say, 4 : 1, and the engine run "all out" at a constant speed. The compression was then gradually increased, and the general running and behaviour of the engine carefully watched. As the compression-ratio rose gradually from 4 to about 4.7, the power output would rise also approximately in proportion to the air standard efficiency, and the running of the engine generally would be smooth and silent, with very little vibration. Between 4.7 and 4.8 the running would become slightly harsher; from 4.8 to 4.9 there would be a very noticeable increase in harshness, accompanied by a considerably greater volume of noise and a marked increase in vibration, while at 5 : 1 the first audible signs of detonation would be detected, though they would be apparent only when ignition was fully advanced and the mixture strength also was adjusted to give maximum possible power.

With a compression-ratio of 5 : 1 the engine would continue to run and develop its maximum power for an indefinite period, but pinking slightly all the time. If the compression-ratio were further raised to about 5.1 : 1 and the ignition and mixture settings unaltered, detonation would increase in severity until, after about 5 or 10 minutes, pre-ignition would set in, and the power would fall off. If, however, the mixture strength were increased to give a slower burning mixture, or the ignition timing retarded, the compression could be raised considerably higher, even to as high as 5.25 : 1 without any perceptible detonation. The same result could also be obtained by reducing the mixture strength, though freedom from detonation would become apparent only when the strength was so far reduced as to leave very little range. The total available range of mixture strength on the weak side is very narrow in the case of all volatile hydrocarbon fuels, so narrow indeed that to attempt to escape detonation by working entirely on the weak side would be completely out of the question in any practical multi-cylinder engine, on account of irregularities in distribution. That it is possible to defer detonation by working between these very narrow limits is only what might be expected from a consideration of the factors controlling detonation. This condition can readily be demonstrated as a laboratory experiment on a single-cylinder engine equipped with very delicate adjustments, and is interesting from a scientific point of view, but it can hardly be considered to be applicable in practice.

So far as practical considerations are concerned, the rate of burning can be reduced, and at the same time stable conditions of running

can be maintained, only by working on the rich side of the range. Under these conditions, however, the power output and economy would cease to rise with increase of compression, because either the time of ignition or the mixture strength would no longer be the most favourable. By further enriching the mixture and retarding the ignition, the compression could be increased to as high as 5.4 : 1, but, of course, only at great sacrifice of economy. The power and consumption curves obtained from such a test are shown in Fig. 8.

4.—Highest Useful Compression-ratio.

It will be observed that, although the first indications of detonation occurred with a compression-ratio of about 4.75 : 1, it is possible by suitable adjustments of the mixture strength and ignition timing to raise the compression to as high as 5.4 : 1 without allowing the detonation to become so severe as to set up pre-ignition. Although the precise compression at which detonation becomes serious varies with different mixture strengths or ignition timing, the compression at which it develops with the most efficient mixture strength and ignition timing is quite clearly defined, and that to within very narrow limits. *This point, namely, the compression at which detonation first becomes audible with certain definite temperature conditions and with both the ignition and mixture strength adjusted to give the highest efficiency, is taken hereafter as representing the highest useful compression-ratio for any fuel. It is not the highest compression which can be employed, but the highest that it is worth while to employ.*

It is necessary to emphasise this definition very clearly because it has a very important bearing upon what follows, and largely explains why certain fuels or mixtures of fuels appear to give more power and greater efficiency than others in the same engine.

The explanation of this is evident enough. Most engines have already too high a compression to enable them to run efficiently on many or even most of the brands of petrol now on the market, with the result that they have to operate with the ignition too far retarded and the mixture too rich. On changing over to another fuel having a higher detonation point, it at once becomes possible to advance the ignition timing and correct the mixture strength, with an immediate gain both in power and economy. Had the same two brands of petrol been tried in an engine with a lower compression-ratio no perceptible difference in power or efficiency would have been observed. It will be shown later that when various fuels are compared at a compression-ratio low enought to permit of the most efficient mixture strength and ignition timing being used in each case, the efficiency is substantially the same on all fuels. Likewise it will be shown that so long as the liquid fuel is completely evaporated before its entry to the cylinder the power output also is substantially the same for any fuel.

5.—Factors Influencing Detonation.

Before dealing with the effect of the composition of the fuel upon its tendency to detonate, it will be well to consider certain other factors which influence detonation.

(*a*) The design of the combustion chamber and the position of the sparking-plug have a very important influence. The more compact the combustion chamber, and the more central the position of the igniter, the less will be the tendency to detonate. For it is clear that with a compact combustion chamber there is less danger of any unburnt portion of the gas becoming stagnant and therefore unable to give up its heat at a greater rate than it is generated by compression due to the burning portion of the working fluid. Again, it is clear that the sparking-plug should be as nearly as possible in the centre of the combustion space, in order to reduce as far as possible

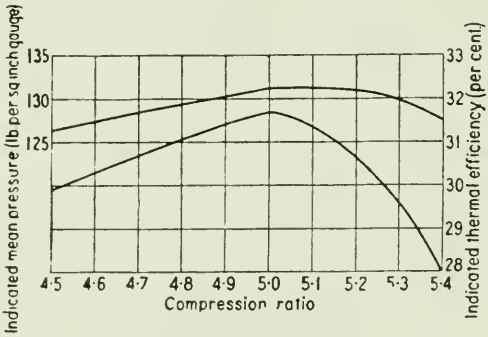


Fig. 8.—Variation in mean effective pressure and thermal efficiency with different compression-ratios for a fuel detonating normally at a compression-ratio of 5.0 : 1.

the distance the flame has to travel. With the exception of sleeve-valve engines, it is usually impracticable to place the sparking-plug in the centre of the combustion space, and the question therefore arises as to the best position for it. Consideration will show that if the gases must be ignited from one side of the combustion chamber, the sparking-plug should be placed on the hottest side, that is to say, on the side nearest the exhaust valve, so that the unburnt gases are compressed against the coolest surfaces, and can therefore get rid of their heat the more rapidly. When the sparking-plug cannot be fitted centrally in the head it is clearly preferable to provide two plugs at opposite sides, so that the spread of flame takes place from two opposite points and has only about half the distance to travel. The use of two plugs is, however, of little value unless the time of passage of the spark across both is perfectly synchronised.

(*b*) *Turbulence.*—The greater the degree of turbulence within the combustion chamber the better, since turbulence tends to spread and distribute the flame uniformly throughout the working fluid and so reduces the danger of portions of it being entrapped and detonated. From the point of view of detonation the worst possible condition is reached when the exhaust valve is fitted in a side pocket in which the mixture may become more or less stagnant, and the igniter fitted at the opposite side of the combustion chamber. In such a design (and it is not an uncommon one) a portion of the working fluid will be entrapped in a position ideal for detonation, where the bulk of the surface to which it can impart its heat is made up of a hot exhaust-valve head and, generally, an uncooled plug for the removal of the valve. Under these conditions detonation will occur very readily, but it may not so readily develop into pre-ignition, because the sparking-plug may have very good facilities for getting rid of its heat, and the points are not so readily overheated by the return wave.

(*c*) *Speed of Rotation.*—It is, of course, common knowledge that the tendency to detonate varies with the speed of rotation, and indeed this is only what might be expected, for the lower the speed, the less the turbulence. As the speed is reduced, not only is the velocity of entrance reduced in the same proportion, but the time during which turbulence may die down is correspondingly prolonged. Again, although the mean temperature of the working fluid at the end of the compression is, no doubt, somewhat lower at low speeds, yet the temperature of the surface of the combustion chamber necessarily varies very considerably, and this, coupled with the lack of turbulence, results in the formation of local zones of gas, in the neighbourhood of the exhaust valve for example, of much higher temperature than the average, and which may in fact attain a temperature approaching very nearly to the self-ignition temperature of the fuel. The existence of these hot zones coupled with a condition of general stagnation naturally predisposes the very conditions favourable to the development of detonation.

(*d*) *Effect of Temperature upon Detonation.*—Since detonation is brought about by the compression of some portion of the working fluid to a temperature in excess of its self-ignition temperature, it follows that for any given normal rate of burning, the higher the temperature of the working fluid the greater will be its tendency to detonate, since the rise of temperature required to produce spontaneous combustion is correspondingly less.

(*e*) *Effect of Pressure upon Detonation.*—Apart from the composition of the fuel, the pressure of compression appears to be by far the most important factor controlling detonation. It appears that the normal rate of flame propagation prior to the formation of an explosion wave bears some direct relation to the pressure of the gases, the explanation presumably being that under high pressure the fuel

and air are in more intimate contact, and therefore each particle of fuel can the more readily find the oxygen necessary for its combustion. It will be shown later that the pressure of compression was found to play relatively a considerably more important part than the temperature of the gases at the end of the compression.*

6.—The Supercharging Engine.

For the investigations into the relationship between the composition of the fuel and the tendency to detonation both the research engines referred to previously were employed. In this connection the supercharging engine working with a stratified charge proved particularly valuable. This engine was, for the purpose of these tests, fitted with a piston giving a normal compression-ratio of 5.18 : 1, corresponding to a compression-pressure of 116 lb. per sq. in. (gauge). When desired, additional air, compressed below the piston, could be admitted to the cylinder after the completion of the suction stroke through ports cut near the bottom of the cylinder barrel, thus raising the compression-pressure to approximately 155 lb. per sq. in. Before entering the cylinders the air—compressed to a pressure of 9 lb. per sq. in.—is forced to pass through an intercooler by means of which it is cooled down to a temperature approximating that of the cylinder contents. It will be seen, therefore, that when the supercharge is admitted by opening the ports in the cylinder wall, the pressure of compression is raised by approximately 35 per cent, but, since the ratio of compression is unaltered, the temperature is unchanged. By this means, therefore, it became possible to raise the compression-pressure without raising its temperature appreciably.

By admitting the air supercharge through ports near the bottom of the cylinder and by suitable formation of the ports and the head of the piston, it was found possible so to separate the active working fluid and the air that they did not completely mix during the compression stroke, and by this means the presence of a considerable excess of air over that required for complete combustion could always be assured, thus permitting what is not obtainable in an ordinary engine, namely, a very wide range of mixture strength over which combustion was complete. In the supercharging engine, therefore, two variables, always present in any normal type of engine, were almost eliminated, namely, the temperature of compression and the mixture strength. Thanks to the absence of these two variable factors it was found possible to arrive at the detonation point of any fuel not only very quickly, but also with a very high degree of accuracy.

* For later results see Proc. N.E. Coast Inst. of E. and S., "The Causes of Detonation in Internal Combustion Engines," by Tizard, May, 1921, and Proc. R. Aeronautical Society, "Thermodynamics of Aero Engines," by Ricardo, Nov., 1924.

To attempt to analyse fully the process of combustion in such an engine is beyond the scope of this report, but it is sufficient to state that this method afforded a means of comparing the detonation points of various fuels under strictly comparable conditions, and with great readiness and fair precision. It is, however, open to the obvious objection that it does not afford any means of expressing the detonation point in terms of compression-ratio, which, of course, is the practical issue: this deficiency was, however, met by means of check tests in the variable-compression engine.

In the case of the variable-compression engine the actual compression-ratio at which detonation first became apparent could be found by gradually raising the compression. Since, however, the tendency to detonate depends largely upon the mixture strength, it was always necessary to ensure, by making fuel-consumption tests, that the correct mixture had been found, and in view of the fact that for every fuel tested a different carburettor adjustment had to be found, it will readily be realised that the detonation point could not be determined so quickly or with quite such precision as in the supercharging engine. Also in the variable-compression engine, probably owing to its more rigid construction, detonation was less easily detected.

7.—Testing Procedure.

The procedure adopted in nearly every case was first to test each fuel against the standard fuel in the supercharging engine, and so determine its detonation point in relation to that of the standard, the detonation point as found on this engine being expressed in terms of the indicated mean pressure at which detonation became apparent. The same samples were then tested in the variable-compression engine and a definite relationship established between the mean pressure at which detonation occurred in the supercharging engine, and the corresponding compression-ratio in the variable-compression engine. With one or two notable exceptions the results tallied reasonably closely, so that once the relative detonation point of any fuel had been determined in the supercharging engine, the highest useful compression-ratio for that fuel could be predicted fairly closely. There were, however, one or two exceptions to this rule, and in the case of these exceptions it must be assumed that the relative influence of temperature and pressure upon the tendency to detonate differed from that of the majority.

In view of the influence both of speed of rotation and of temperature upon the tendency to detonate, great care was taken to ensure that the same temperature conditions prevailed in both engines and that the speed of rotation was always the same.

In all, many hundreds of tests of the nature described above were carried out. Not only were samples of practically all the brands of commercial petrol and benzol tested but also nearly pure samples

of several members of each of the series comprised in petrol including various sulphur compounds, also alcohol, acetone, acetylene, ether, etc., were investigated.

8.—Detonation with Various Fuels.

In the case of ethyl-alcohol, acetone, toluene and xylene, it was found quite impossible to induce detonation under any circumstances in either engine. The relative detonation point, however, of these fuels could be determined by preparing mixtures of each of them with the standard aromatic free petrol or with hexane. It was found that, in all cases, the detonation point of such mixtures followed a straight law when the mixture was apportioned by weight and not by volume; that is to say, the addition of 40 per cent by weight of, say, toluene to hexane would raise the detonation point exactly four times as much as the addition of 10 per cent. One interesting point was repeatedly observed. When benzene and certain other fuels also were tested alone in the variable-compression engine and attempts made to induce detonation by raising the compression, occasional pre-ignition would occur when the compression-ratio exceeded 7 : 1, these pre-ignitions being denoted by a dull thud and an immediate drop in speed and power. It would appear, therefore, that such fuels, when used pure, pre-ignite without preliminary detonation, but the addition of a comparatively small proportion of paraffins would induce preliminary detonation at high compression-ratios. The same applies also to the naphthenes, methyl alcohol, methylated spirits, and one or two others.

Table II., p. 144, gives the detonation point as found in both the variable-compression and supercharging engines.

In Fig. 9 is shown the variation in mean pressure and efficiency with compression-ratio as found by actual experiments on the variable-compression engine. These curves represent the mean of a great number of tests carried out on a large number of different fuels, and since in no case did the results vary by more than 1 per cent the accuracy of the mean result may be accepted as being of a high order. It is very important to note that the relationship between power output (expressed in terms of indicated mean pressure) and economy (expressed in terms of indicated thermal efficiency), and compression-ratio was found to be the same for all fuels. Examination of these results will reveal that the inter-relationship between power output and thermal efficiency varies at different compressions. This can be partially, but not, in the author's opinion, wholly, explained by certain investigations which will be dealt with later on.

9.—Efficiency, Power and Toluene Value.

In Table II. are given the efficiency and power output obtainable from many different relatively pure samples, and also a number of

composite fuels. As already explained, it is improbable that it will ever be worth while to employ a compression-ratio in excess of 7 : 1, and the efficiency corresponding to this compression-ratio has been taken as 100. It is well to bear in mind, however, that toluene, xylene, alcohol, and one or two others will, in fact, withstand a

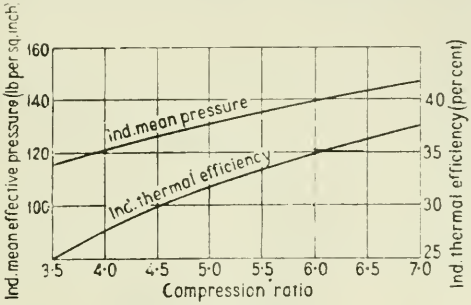


FIG. 9.—Variation in indicated M.E.P. and indicated thermal efficiency with different compression-ratios for a fuel not detonating below a ratio of 7 : 1 (say for toluene) in the variable-compression engine.

considerably higher compression without detonation. Prior to the investigations on ethyl alcohol, and the members of its group, the highest detonation point observed was that of toluene, and it was therefore decided, for the sake of convenience, to express the tendency

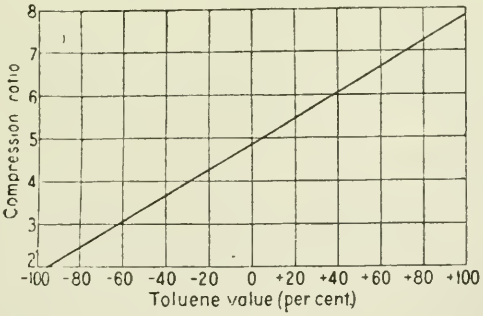


FIG. 10.—Relation between toluene value and compression-ratio in the variable-compression engine.

of fuels to detonate in terms of their equivalent toluene content. This value will in future be termed the "toluene value" of the fuel.

With the exception of alcohol and the members of its group, the efficiency obtainable with all fuels appears to be the same at the

same compression-ratio to within less than ± 0.5 per cent. Since the toluene value defines the compression-ratio which may be used, it forms a definite and precise measure of the efficiency obtainable from any fuel when used at its most efficient compression-ratio. The curve in Fig. 10 shows the observed relationship between toluene value and compression-ratio, and Fig. 11 the observed relationship

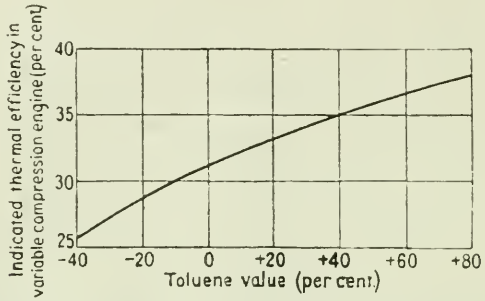


FIG. 11.—Relation between toluene value and indicated thermal efficiency in variable-compression engine.

between toluene value and efficiency, while Fig. 12 shows the relationship between toluene value and indicated mean effective pressure or power output. The curves shown in Figs. 10 and 11 are strictly applicable to all fuels outside the alcohol group, and with but very slight amendment to that group also. The relationship between toluene value and indicated mean pressure assumes that all

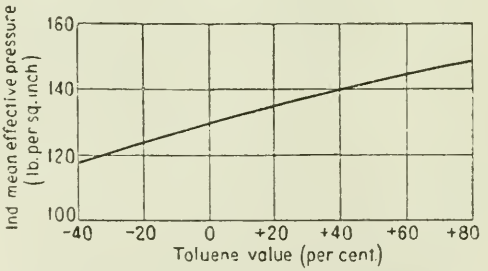


FIG. 12.—Relation between toluene value and indicated mean effective pressure in variable-compression engine.

fuels have the same total internal energy per unit volume of mixture, which is very nearly, but not strictly, true. This subject will, however, be dealt with when discussing the maximum power output obtainable from different fuels.

Table II. also gives the toluene value for a large number of different samples and composite fuels. Reference to this Table will show

that, of the aromatic group, benzene is markedly inferior to toluene in checking detonation, while xylene occupies a position intermediate between the two. Ethyl alcohol is by far the most effective of all. With the exception of ether and acetylene, the members of the paraffin series are the worst, and, generally speaking, their tendency to detonate increases with increase of molecular weight. In the estimates of toluene values, the standard aromatic free petrol is taken as zero, and toluene as 100 per cent.

Possibly it would have been better to have selected either a pure paraffin fraction or a range of pure paraffins as a zero standard, but neither of these were procurable either in sufficient quantity or of a sufficient degree of purity for the purpose. It must be borne in mind when making comparisons that, although the aromatic free petrol contains less than 2 per cent of aromatics, it does, nevertheless, contain a substantial proportion—about 35 per cent—of naphthenes, and that these have a positive toluene value of about 25 per cent. If a fuel consisting of a similar range of paraffins only had been selected as forming a datum line, then the aromatic free petrol would have a positive toluene value of about 10 per cent. The standard aromatic free petrol was selected because it was available in very large quantities, and, the whole bulk of it having been prepared at one operation, its consistency could be relied upon as being uniform —a very important factor in the case of a fuel used throughout as a standard basis for comparison.

Ether and acetylene, and the heavier paraffins, have, therefore, minus values, while hexane (a very light paraffin) has a very slightly positive value. In the case of the paraffin series it was found difficult to determine the detonation point of the heavier members of the group, partly because these could not be isolated in sufficient quantities or of a sufficient degree of purity for the purpose, and partly because a considerable amount of heat had to be added in order sufficiently to vaporise them.

There was, however, considerable evidence to show that the toluene value falls as the molecular weight increases. Small quantities of heptane of a very high degree of purity were eventually obtained, and it is hoped that sufficient supplies of both octane and nonane may eventually be procured. The results obtained on heptane were as interesting as they were surprising, for, contrary to all expectations, not only did this fuel detonate far more readily than the aromatic free petrol of which it forms one of the relatively lighter constituents, but when used pure it detonated with a compression-ratio of only 3.75 : 1, and when mixed with hexane, even in quite small quantities, it increased the tendency of the latter to detonate.

Experiments on the variable-compression engine showed that heptane had a negative toluene value of minus 37 per cent. The particular sample of heptane with which the experiments were conducted was of a vegetable origin. Before any tests were run it was

very carefully analysed, and, but for a small proportion, about 3 per cent of turpenes, was found to be remarkably pure. The results obtained were so unexpected that suspicions were aroused as to the possible influence of the presence of turpenes. Experiments with mixtures of petrol and turpentine, however, showed that the addition of turpentine served to reduce the tendency to detonate. Finally, the remainder of the heptane was eventually almost completely freed from turpenes, and its purity raised to over 99.5 per cent; but, even so, its behaviour was not influenced to any perceptible degree.

The quite remarkable difference observed between the behaviour of hexane and heptane, two contiguous members of the same series, emphasises very strongly the necessity for taking nothing for granted, and for investigating, individually, the behaviour of each member of a series; moreover, there is reason to believe that certain other members of the paraffin series behave in the same erratic manner.

In this connection it may be of interest to note that a sample of commercial kerosene, consisting mainly of heavy paraffin fractions, detonated at a compression-ratio of 4.2 : 1. This is equivalent to a toluene value of minus 22 per cent. Under the same conditions of heating the toluene value of hexane was found to be plus 8 per cent and of pure heptane to be minus 37 per cent. However, in spite of the above, and possibly other notable exceptions, there is some evidence to show that as a general rule the heavier members of the paraffin series detonate more readily than the light ones. Applied to this series, and to this series alone, the accepted specification, based on boiling points, affords a very approximate indication of the quality of the fuel; but when only a small percentage of aromatics or even naphthenes is present, the position is entirely reversed, and the whole basis of the accepted specification at once falls to the ground.

1.—Relative Importance of Compression-Temperature and Pressure.

In order to establish definitely the relative importance of temperature and pressure at the end of compression upon the tendency to detonate, hexane was evaporated in a small steam boiler and admitted to the carburettor. The temperature of the entering gases was then controlled by means of the electric heater fitted in the carburettor air intake passage. When working in this manner it was found that the compression-ratio could be raised by 0.1 for every 10 degrees C. drop in temperature from 70° C. to 30° C., the corresponding ratios being from 4.8 : 1 up to 5.2 : 1.

This test is important, because it establishes a definite relationship between the temperature of the charge and its tendency to detonate.

Nor is the issue confused by any question of a further drop in temperature after entry to the cylinder due to the latent heat of unevaporated liquid. So long as no evaporation takes place within the cylinder the temperature of compression at any given speed—and cylinder temperature will always bear a definite relationship to the temperature of the entering fluid over the range of compression-ratio explored—the final compression (absolute)-temperature may be taken as approximately 1.7 times the absolute temperature at the commencement.

Thus an increase in the temperature of compression of 17 degrees C. necessitated a drop in the compression-ratio of 0.1.

If, as is generally supposed, detonation were a function of temperature alone, then the variation in compression-ratio corresponding to the variation in inlet temperature would have been nearly five times as great, for the difference between the final compression-temperatures at compression-ratios of 5 : 1 and of 6 : 1 is only 38 degrees C. if the inlet temperature is constant.

These direct, and other less direct, experiments all confirm the statement made previously that the detonation point of a fuel is more closely dependent upon compression-pressure than upon compression-temperature.

2.—Relationship Between Detonation Point and Spontaneous-ignition Temperature.

It has generally been assumed that the detonation point of a fuel would be dependent upon its spontaneous-ignition temperature as determined in air or oxygen. While, broadly speaking, the results showed that the temperature of spontaneous ignition of a fuel might be taken as a very approximate indication of its tendency to detonate, it was found to be quite unsafe to rely upon this relationship.

Table II. shows the compression-ratio at which certain samples were found just to detonate. The temperature of spontaneous ignition, in air and oxygen at atmospheric pressure, as determined by Mr. Harold Moore, is shown in Table IV., p. 148.

If detonation were a function of compression-temperature alone, then a closer relationship might be looked for between the temperature at which detonation commenced and the temperature of self-ignition as determined by Moore's apparatus. In any event, however, the conditions obtaining in an engine cylinder, in which the gases are in a state of violent turbulence, are very different from those under which the tests for spontaneous ignition were carried out.

It seemed probable that the temperature of spontaneous ignition of a fuel would form a tolerably true indication of its readiness to pre-ignite, as distinct from the tendency to detonate. This seemed to be borne out in the case of those fuels which pre-ignite without

preliminary detonation, such as the members of the aromatic series, methyl alcohol, etc., but the following experiments with mixtures of the paraffin series and carbon bisulphide tended to corroborate this theory. Experiments on the self-ignition temperature of carbon bisulphide when ignited as a stagnant mixture by adiabatic compression showed that this fuel would ignite spontaneously at a temperature of about 275° C. When this fuel was tested alone in the variable-compression engine heavy pre-ignition occurred at the lowest compression-ratio, and at the lowest inlet and water temperature, with the result that it could not be used at all.

The pre-ignition was, however, unaccompanied by any trace of detonation. When carbon bisulphide was mixed with aromatic free petrol in proportions up to and even exceeding 50 per cent, it was found to raise the detonation point as compared with that of the petrol alone. A mixture of aromatic free petrol and carbon bisulphide in equal parts was found to detonate with a compression-ratio of 5.15 : 1, while pure aromatic free petrol detonated at a compression-ratio of 4.85 : 1. So far, therefore, as detonation is concerned, carbon bisulphide may be said to have a positive toluene value of about 18 per cent. When running with such a mixture at a compression-ratio of 5.15 : 1 there was no trace of pre-ignition, and it was not until the proportion of aromatic free petrol was still further reduced that pre-ignition first became apparent.

The inference to be drawn from these experiments is that carbon bisulphide, while pre-igniting very readily, will not detonate, whereas any paraffin petrol will detonate, but will not readily pre-ignite. Tests carried out on the special machine designed for ascertaining the self-ignition temperature of fuels when ignited by adiabatic compression showed that carbon bisulphide ignited under compression at a gauge pressure of about 185 lb. per sq. in. and a compression-temperature of about 275° C., while the aromatic free petrol ignited under the same conditions at a gauge pressure of 320 lb. per sq. in. and at a temperature of 353° C. The initial temperatures before compression were 15.0° C. and 15.6° C. respectively.

In the case of those fuels which pre-ignite without preliminary detonation, it was found very difficult indeed to determine the pre-ignition point, because, in practice, this clearly depends largely upon the temperature of the exhaust valves, sparking-plug electrodes and other insulated parts within the combustion chamber. As might be expected, variations in the design, or even in the condition of the sparking-plug, exerted so great an influence upon the tendency of the fuel to pre-ignite as to render anything more than a very approximate determination impossible.

It is hoped, later on*, that the results of tests of the spontaneous ignition temperature and pressure, when both temperature and

* See "The Causes of Detonation in Internal Combustion Engines," by Tizard, *loc. cit.*

pressure are raised by the adiabatic compression of a stagnant mixture, will be sufficiently advanced to enable them to be fully dealt with, and their relation to the detonation point shown.

It may be argued from the test results given that so far as existing engines are concerned, no advantage would be obtained from the use of a fuel of high toluene value, because very few engines, other than aircraft, have a compression-ratio as high as 5 : 1, which in the variable-compression engine requires a toluene value of only 5 per cent.

It must be remembered, however, that in this engine every effort has been made to reduce detonation. The form of the combustion space is the most efficient practically obtainable; the charge is ignited from two points at opposite sides of the combustion chamber, and the time of the passage of the spark perfectly synchronised; turbulence of the charge in the cylinder is at a maximum, and there are no pockets or recesses where unburnt gas can become entrapped and detonated.

Finally, all comparative tests were run at a speed of 1500 revs. per minute, corresponding to a piston speed of 2000ft. per minute.

3.—Toluene Value Required for Commercial Spirit.

In the ordinary car or commercial-vehicle engine these ideal conditions do not apply, but, on the contrary, the combustion chambers are in many cases so designed as to encourage detonation. Check tests were from time to time made on various standard engines on different test beds at the laboratory. One engine, designed and built by one of the best-known makers of commercial-vehicle engines, afforded an excellent illustration of the effect of combustion-chamber design upon detonation. Had the designer of this engine set himself the task of trying to induce detonation, he could hardly have been more successful. This particular engine had a compression-ratio of 3.95 : 1, corresponding to a compression-pressure of only 78 lb. per sq. in., yet the engine detonated alarmingly even on a fuel of toluene value 10 per cent, while the variable-compression engine using the same fuel would run perfectly smoothly, and at maximum efficiency, with a compression-ratio of 5 : 1. To enable the particular engine referred to to run smoothly on any fuel having a toluene value of less than 20 per cent, it was necessary to employ an over-rich mixture and considerably to retard the ignition, thus sacrificing both power and economy.

It is estimated that the majority of engines now in use can run at their maximum efficiency on a fuel of toluene value of from 15 per cent to 20 per cent.

In the summer of 1919 the Shell Marketing Co., realising that the existing methods of specifying the quality of their motor spirit by determination of the specific gravity and boiling-point range, while

affording useful information in certain respects, were of minor importance compared with toluene value, inagurated the system of so blending their Shell spirit as to give a high uniform toluene value. During this period a census was taken of the experience of fifty users of a total number of nearly 200 cars and commercial vehicles. An analysis of this census showed an average increase in the mileage per gallon of no less than 17 per cent as compared with other brands of petrol, while over 80 per cent of the users canvassed volunteered that they noticed a very marked increase in power on hills, due clearly to the fact that absence of detonation enabled them to run at slow speeds with an efficient ignition timing. Guided by these results, the Shell Marketing Co. have determined to maintain a definite toluene value standard for their Shell spirit, and they now always blend the brand with this object in view.

This practical experiment proved, strikingly, that it is the tendency of a fuel to detonate which in practice controls the efficiency and power output obtainable from it, even when it is used in the relatively low-compression engines such as are general to-day.

Tests carried out by the Royal Aircraft Establishment and most of the manufacturers of aero-engines have shown also that both the power output and economy of aero-engines can be increased by the addition of benzol to fuels of low toluene value, not because it contains greater internal or potential energy, for, as will be shown later, it does not, but simply because the addition of benzol checks the tendency to detonate and so permits of the more efficient use of the fuel. The Air Ministry's existing* specification for an aircraft spirit insists upon a specific gravity and boiling-point range which limits the toluene value to from 12 to 14 per cent, and necessitates a low aromatic content for the fuel. Such a fuel will cause violent detonation in some of the most important existing aircraft engines. The aromatic content therefore is often raised by the addition of benzol, which is composed chiefly of benzene and which, curiously enough, is the heaviest and least effective member of the aromatic group. Such a state of affairs illustrates how little the essential characteristics of a good fuel were understood.

It will be shown later that, compared with the question of detonation, all other considerations and differences, in so far as they control the power output and efficiency, sink into insignificance.

Certain rigid requirements must, of course, be fulfilled. The fuel must contain at least a minimum proportion of highly volatile constituents with a high vapour tension in order to facilitate starting; these, however, can be added to any fuel, and need not influence appreciably its toluene value. Again, the final boiling point must not be so high that liquid fuel will condense upon the cylinder walls, and so ultimately find its way into the crank case. These and other

* This statement no longer applies.

points will be dealt with later on, but it is well to emphasise at this stage that, compared with the tendency of a fuel to detonate, they are of secondary importance.

4.—Use of Inert Gases.

In the course of the investigations upon the subject of detonation certain other experiments were tried, some of which are of sufficient interest to be worth recording.

Since detonation is presumably a function of the rate of burning of a fuel, experiments were carried out with a view to ascertaining how far the efficiency could be improved, when the rate of burning was controlled by means other than varying the composition of the fuel. For this purpose the addition both of cooled exhaust gases and of water was employed.

It was found that by the addition of cooled exhaust gases to the standard aromatic free petrol, detonating normally at a compression-ratio of 4.85 : 1, the compression could be raised up to 7.5 : 1, or even higher, without detonation or pre-ignition. This increase of ratio corresponded to an increase of compression-pressure of from 105 lb. per sq. in. to 203 lb. per sq. in. gauge. In these tests the variable-compression engine was run up on the standard aromatic free petrol, until all conditions had become normal. The compression-ratio was then raised till the detonation point was reached, when readings of torque and fuel consumption were taken, the mixture strength being adjusted until the most economical proportions had been found. A cock admitting cooled exhaust products to the air intake of the carburettor was then slightly opened and the compression raised until detonation again became apparent, when readings of torque and fuel consumption were taken at the higher compression-ratio. The procedure was repeated at frequent intervals at compression-ratios ranging up to 7.5 : 1, in each case just sufficient exhaust gas being admitted to check detonation. The results obtained are shown in Fig. 13, in which the full lines show the torque in terms of indicated mean pressure and the dotted lines show the torque obtained without exhaust gas added, at the same compression-ratios and with a fuel of the same energy content, but of sufficiently high toluene value to withstand the highest compression. The divergence between the two mean-pressure curves indicates approximately the proportion of exhaust products required to check detonation at each compression-ratio for aromatic free petrol. The lower curves show, in full lines, the fuel consumption in pints per indicated horse-power per hour, with the exhaust gas added, and in dotted lines, the consumption obtained with a fuel of exactly the same calorific value per pint, but of sufficiently high toluene value not to need the addition of exhaust gas.

It will be observed that in spite of the addition of inert gases, the

net power increased with the increase of compression up to a ratio of 5.5 : 1. This is due clearly to the higher efficiency of the "air standard." Above 5.5 : 1 the quantity of inert gas required to check detonation more than balanced the improved efficiency due to the greater expansion ratio, and the power output declined again.

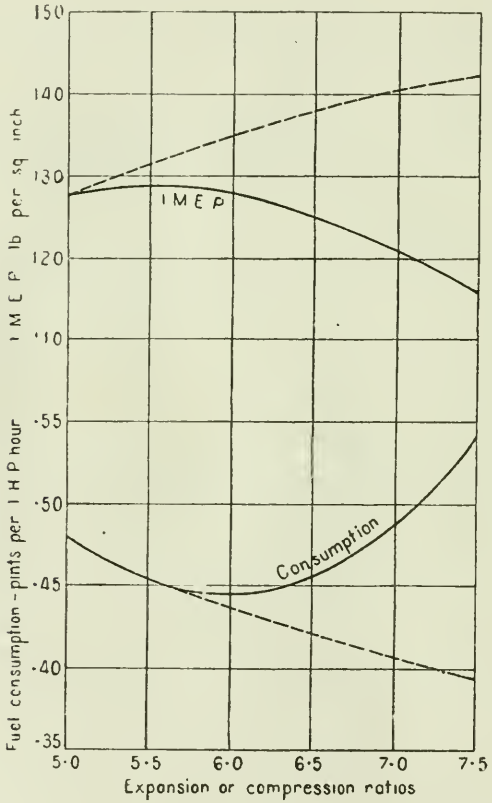


FIG. 13.—Full-line curves show the results with a fuel of low toluene value with cooled exhaust gas added. Dotted lines show the corresponding values of I.M.E.P. and of fuel consumption for a fuel of sufficiently high toluene value to require no dilution with inert gases.

The economy increased with increase of compression up to 6 : 1, in spite of the high specific heat of the exhaust products, but above this compression it declined rather rapidly, due in part to the rapidly increasing proportion of exhaust products required, and in part no doubt to incomplete combustion.

Similar tests were carried out, using steam in place of cooled exhaust gases. The results obtained were generally similar in characteristic. It is hoped* to carry out further tests with nitrogen and carbon dioxide.

The results given are interesting in that they indicate clearly :—

(*a*) That under normal conditions pre-ignition is, in the case of ordinary petrols containing a substantial proportion of members of the paraffin series, always brought about by persistent detonation.

(*b*) That if detonation be prevented, pre-ignition will not readily occur, even though the compression-pressure be raised to double that permissible under normal conditions.

(*c*) That detonation is a function of the normal rate of burning, and that if this be delayed by employing inert gas to act both as a brake and as a cooling agent, much higher compression-pressures can be used.

5.—Experiments with Compression Varied by Throttling.

Another experiment, also of some interest, was carried out in order to determine the compression-pressure at which detonation takes place when a high compression is used and the engine is throttled.

For these tests the same standard aromatic free petrol was used, and after preliminary running at full throttle the throttle was partially closed, and the compression raised to, say, 5.5 : 1. The throttle was then very gradually opened until detonation just became apparent, a reading of torque was taken, the throttle was locked securely in position, and the supply of fuel cut off. The engine was then motored at the same speed, and the actual compression-pressure recorded by means of a modified form of Okill indicator, which was found to give very accurate readings. The same procedure was repeated at a number of different compression-ratios ranging up to 7.5 : 1, and the results obtained are as shown in Fig. 14.

The full lines indicate the observed results with varying throttle openings, and the dotted lines show what would have resulted if the fuel had permitted the use of full-open throttle throughout. It will be observed that, while under these conditions of throttling the indicated mean pressure, and therefore the power output, naturally falls very rapidly, the actual compression-pressure at which detonation first became apparent is nearly constant. However, owing to the relatively small proportion of charge taken in at the lower throttle openings, the proportion of exhaust gases in the mixture is slightly greater at the higher compression, and this no doubt accounts for the fact that the pressure at which detonation commenced is higher at the higher compression-ratios instead of slightly lower, as would

* See "Causes of Detonation," Tizard, *loc. cit.*

otherwise be expected, due to the somewhat higher compression-temperature.

6.—Aniline Test and Toluene Value.

As a practical and ready means of determining the toluene value of a fuel it is interesting to note that the new aniline test employed by chemists for determining the aromatic content of a fuel responds also to the naphthene content, and therefore this test affords a fairly

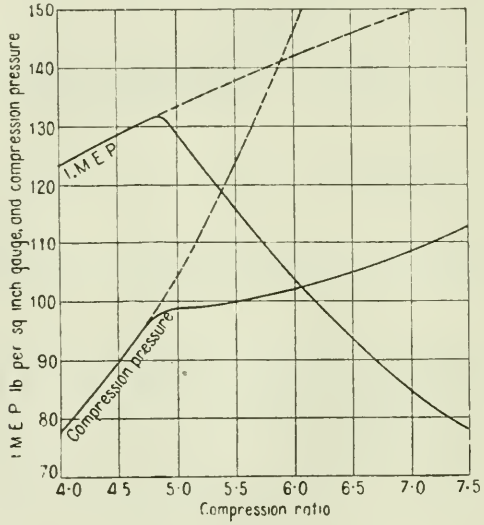


Fig. 14.

The full lines show the indicated mean pressure obtained and the corresponding compression-pressure (taken by modified Okill gauge) for various compression-ratios, and the throttle closed just sufficiently to prevent detonation of a fuel of low toluene value. The dotted line continuations show what would have been the resulting mean pressures at the various ratios had the toluene value been sufficiently high to admit of full-open throttle throughout.

accurate measure of the tendency of the fuel to detonate. The reason for this is shown by reference to Table II, which gives the observed toluene values and approximate compositions of various fuels. It may be seen from this Table that the average toluene value of the paraffins present in petrol is roughly − 10 per cent, the average for the naphthenes is + 30 per cent, the average for the aromatics is + 75 per cent, the toluene value of benzene being 67 per cent, of toluene 100 per cent, and of xylene 85 per cent.

Althought the results obtained from this series of investigations into the factors controlling detonation permit of certain fairly definite conclusions being drawn, the author feels that much still

remains to be done, for many discrepancies still remain to be cleared up. The following conclusions and inferences may, however, safely be drawn :—

(*a*) The one outstanding factor, limiting both the power output and the efficiency obtainable from any fuel, is the tendency of the latter to detonate. This controls not only the compression-ratio which may be employed, but in most cases it controls also the efficiency at which the fuel can be burnt, even at relatively low compressions.

(*b*) The tendency of a fuel to detonate appears to be a direct function of its normal rate of burning.

(*c*) No evidence of any kind has been adduced to show that the normal rate of burning of any fuel can be too low. This point will be discussed further when dealing with the power and efficiency obtainable.

(*d*) Of all the constituents of natural petrol the most effective in checking detonation is toluene, followed by the other members of the aromatic group.

(*e*) Although compared with the other constituents the aromatic series have a somewhat lower calorific value per lb. (but not per gallon), the gain in efficiency due to their presence, even in quite small quantities, is so large as to render the small difference in heat value insignificant.

(*f*) The classification of a fuel by its specific gravity and boiling point applies only when the fuel is composed entirely of pure paraffins. No such fuel is, or ever has been, available outside a chemical laboratory.

(*g*) Any specification for a fuel should be so drawn as to encourage the inclusion of as large a proportion as possible of the aromatic series naturally present in the crude petroleum.

Power Output Obtainable from Different Fuels.

The question of the power output obtainable from different fuels must be regarded from two points of view :—

(1) When all the fuels tested are used with the same compression-ratio.

(2) When the compression is adjusted to suit each fuel, the ratio of compression used being the highest consistent with freedom from detonation in any normal circumstances.

We will deal first with the results obtained at a constant compression-ratio. Under these conditions, provided that each fuel is burned at the same efficiency (and it will be shown later that this is the case), the power output will clearly be proportional to the energy content of the fuel-air mixture. This energy content will be equal to the product of the heat of combustion of the fuel, when burnt with the requisite quantity of air, and the change in specific

volume. By change in specific volume is meant the increase or decrease of the number of molecules in the working fluid during combustion. It is clearly necessary to take account of this in calculating the total internal energy of the burnt fuel-air mixture, since at the same temperature the pressure will be proportional to the number of molecules.

1.—Heat of Combustion of a Fuel-air Mixture.

For a fuel-air mixture the heat of combustion is probably best expressed in terms of ft.-lb. of energy liberated by the combustion of 1 cu. in. of a mixture of fuel and air measured at standard temperature and pressure, the mixture being in all cases of such proportions that all the oxygen in the air is combined.

As pointed out previously, the heat of combustion of a fuel-air mixture as defined above has no direct relation to the calorific value of the fuel.

Table III., p. 146, gives the calorific value in terms of weight and volume, also the heat of combustion of a number of different hydrocarbons previously mentioned, and some other combustible substances.

From this Table the heat of combustion of almost any volatile liquid fuel can be arrived at, once its chemical composition has been determined.

2.—Change of Specific Volume.

The change in specific volume varies, of course, with the composition of the fuel; it varies also with the mixture strength. In the case of all fuels belonging to the general group known as petrol, the specific volume after combustion is substantially greater than before, thus giving a net increase in the total internal energy. In the case of benzene and other members of the aromatic group there is only a slight increase in specific volume, while in the case of hydrogen, carbon disulphide, acetylene, etc., there is a substantial decrease.

Table III., Column E, gives the total energy liberated by combustion in terms of ft.-lb. per cu. in., taking into account the change of specific volume, and assuming that the proportions of the mixture are such as to give complete combustion. The figures given in this column may therefore be taken as an expression of the power output obtainable from each fuel, assuming —

(*a*) That the mixture strength is that giving complete combustion.

(*b*) That the fuel is completely vaporised before entering the cylinder.

Note.—This will only be true, strictly speaking, if Column E of Table III. is calculated using calorific values of the fuels burnt in a state of vapour.

Bomb calorimeter experiments give calorific values from a liquid state, therefore to the usual calorimetric figures should be added the latent heats of evaporation. Strictly speaking, should be added only the latent heat *at constant volume*, which is the usual latent heat (at constant pressure) less $p\,x$ (expansion on vaporisation). This latent heat is apparently about $^{14}/_{15}$ of the figure at constant pressure. Latent heats are almost certainly observed at atmospheric pressure, so that the amount to be subtracted will be simply $\{(14.7 \times 144)/778\} \times$ (specific volume of vapour at normal B.P.).

(*c*) That the temperature of the mixture after complete evaporation is the same for each fuel.

When an over-rich mixture is used, and the fuel is only partially burnt the heat of combustion of the mixture will be no greater, but the increase in specific volume will in most cases be greater, and therefore the total internal energy will be somewhat greater on this score ; therefore a slight increase in power might be expected in the case of those fuels having a relatively large increase in specific volume. On the other hand, when an over-rich mixture is used the products of combustion so formed have a higher specific heat, and this, in nearly every case, counteracts the advantage which would otherwise be obtained from the increase in specific volume. In practice, with, perhaps, the single exception of the alcohol group, very little, if any, increase in power is obtained by using a mixture richer than that required for complete combustion.

An examination of Column E of Table III. reveals the very interesting fact, confirmed by numberless experiments, that the power output available from all known volatile hydrocarbon fuels is the same to within narrow limits provided (*a*) that they are all tested at a compression-ratio at which full ignition advance can be used without detonation, and (*b*) that the losses due to change in specific heat and to dissociation at high temperatures are the same in all cases. In this connection Messrs. Tizard and Pye's investigations prove that there is very little to choose in this respect between the combustible mixtures of any of the available fuels.

3.—Experimental Results.

The experimental results, a *résumé* of which will be given later, and which cover a very large number of independent tests, have completely borne out the conclusions arrived at from Messrs. Tizard and Pye's theoretical investigations* of the chemical properties of the fuels and of the process of combustion. All the experiments have proved conclusively that, so long as the fuel is completely evaporated, the maximum power obtainable from samples of each of the groups given in the above Tables, or from any combination of them, is proportional to the energy content, and, excepting only the alcohol group, is the same to within less than 3 per cent. The differences

* *Ante.*

in the losses due to dissociation and to change of specific heat that do exist are such as to minimise the differences between the heats of combustion of the different types of fuels (paraffins, naphthenes, aromatics, etc.), and to place them still more on a par with one another.

4.—Influence of Latent Heat of Evaporation.

There is, however, one variable factor which has been mentioned previously, and which, so far as the author is aware, has not been taken into consideration by other experimenters. This is the influence of the latent heat of evaporation upon volumetric efficiency. This factor was found, in the case of alcohol and the other members of this group, to have a very powerful influence on the maximum power output obtainable.

The conclusions given so far are based on the assumption that the whole of the fuel is evaporated before its entry to the cylinder, but in practice this is seldom, if ever, the case. It is, of course, clear that the weight of charge taken into the cylinder will be inversely proportional to its absolute temperature at the moment when the inlet valve closes. There is definite evidence from experimental results that, with the exception of alcohol and the other members of its group, all fuels boiling below 200° C. are completely evaporated, before the commencement of the compression stroke, by contact with the hot walls and admixture with the highly-heated residual exhaust products in the cylinder, excepting only a very small proportion which may enter the cylinder in coarse drops, and so not only escape evaporation, but even, to a large extent, combustion also. This proportion is, however, quite insignificant, and has no influence so far as power output is concerned.

The absolute temperature, therefore, at the commencement of the compression stroke is dependent upon (*a*) the amount of external heating applied, and (*b*) the latent heat of evaporation. It is almost independent of the temperature of the charge during its entry to the cylinder. In fact, the final absolute temperature, and, therefore, the weight of the charge taken into the cylinder is dependent upon the quantity and latent heat of the fuel, and upon the amount of heat added to it, external to the cylinder—it is substantially independent of the volatility of the fuel and its entering temperature.

For example, a highly volatile fuel entering the cylinder at 0° C., and a heavy fuel of low vapour tension entering at 30° C., will both have the same final absolute temperature at the commencement of compression, if the latent heat of both is the same, and if both charges received the same amount of pre-heating. In the former case most of the evaporation has taken place outside the cylinder, and the added heat has been absorbed by the latent heat of the vapour. In the latter case little or no evaporation has taken place outside the

cylinder, and the added heat has therefore raised the temperature of the air and the still liquid fuel. In each case contact and admixture with the highly-heated exhaust products in the cylinder will complete evaporation, and in each case the final temperature will be the same; hence the weight of working fluid (which is inversely proportional to the absolute temperature), and therefore the power output, will be the same in each case.

From the above considerations it will be seen that, with any given amount of pre-heating (provided it is not excessive), the volumetric efficiency, and therefore the power output, will be directly proportional to the latent heat of the fuel.

The author has deemed it advisable to emphasise this point very strongly, because he was at first misled into supposing that the final suction temperature was proportional to the inlet temperature, as it would be in the case of a gas engine.

Apart from the alcohol group, the variation in latent heat is not very large, and does not exercise any appreciable influence. It is interesting to note, however, that in cases where the total internal energy is lower, the latent heat is generally slightly higher; consequently a slightly greater weight of charge is taken into the cylinder, sufficient in most cases to compensate for the lower internal energy, and thus bring the actual power output to substantially the same amount in all cases. This point is well illustrated by the instance of benzene, as will be seen at once by reference to Table III. The energy liberated by combustion of a cubic inch (at standard temperature and pressure) of benzene-air mixture is appreciably lower than that of the hydrocarbons forming the greater proportion of petrols. On the other hand, the latent heat of benzene is considerably greater, and as a result the power output obtainable from benzene is the same as in the case of petrol to within less than half of one per cent.

Table I. gives the latent heat of evaporation of a number of hydrocarbons and other substances. The air/fuel ratio by weight, also the drop in temperature of the mixture due to evaporation of the liquid, is shown for each fuel in Table III. The calculations are made for mixtures giving complete combustion, but without excess of air.

In the case of alcohol, owing to the very much higher latent heat and to the fact that the proportion of fuel to air is also much greater, the latent heat of evaporation plays a supremely important part, and results in a really marked increase in power as compared with other fuels, although its total internal energy is lower than that of either petrol or benzol. Moreover, the power output increases very considerably when an over-rich mixture is used, a feature not observed to any marked extent with other fuels. With an over-rich alcohol mixture, more fuel is evaporated, the temperature of the charge is lowered, and the gain in weight of charge or volumetric efficiency considerably more than outweighs the loss due to the greater specific heat of the products of combustion.

Apart from the variations due to the differences in the latent heat of evaporation of fuels, which only assumes important proportions in the case of alcohol and the other members of that group, all tests proved that the power output obtainable was, in fact, directly proportional to the total internal energy of the fuel, and this was further confirmed by the fact that the thermal efficiency at any given compression-ratio was found to be the same with all fuels boiling below 200° C., except the alcohol group. That is to say, it was found that each fuel was burnt at the same efficiency, and, therefore, the same proportion of its internal energy was converted into power. In the case of the alcohol group the efficiency was found to be slightly higher, and a further increase in power output was obtained thereby.

There was clear evidence that in the case of the alcohols, owing to the very high latent heat, evaporation continued throughout the compression stroke also, with the result that the compression was nearly isothermal. As a result the temperature of the cycle was lowered, the losses due to conduction and to change of specific heat were reduced, and the efficiency was slightly increased thereby. This feature became particularly prominent in the case of methyl alcohol, which has the highest latent heat of any of the fuels tested. Since it has also the lowest calorific value it follows that the weight of fuel per cycle is greater than with any of the others. Both the power output and efficiency obtainable with methyl alcohol were considerably greater than with any other fuel examined.

In referring to alcohol and allied substances as fuels it should perhaps be explained that though these fuels give a considerably greater maximum power output, and, furthermore, can be used with a higher thermodynamic efficiency than either petrol or benzol, the heat value of the alcohols per unit of weight or per unit of volume of the liquid is so much lower than that of petrol or benzol that the rate of consumption for equal power output is nevertheless greater. The exact amount of this larger fuel consumption with alcohol, as compared with petrol or benzol, will be examined later when dealing with the subject of efficiency.

In Table V., p. 149, the values given in Column B show the relative power output calculated directly from the foregoing Tables and in accordance with the argument adduced above in arriving at the conditions governing power output. The greatest difference among the paraffins, aromatics, and naphthenes is hardly more than plus or minus half of one per cent. This deduction, as will be seen later, is corroborated almost exactly by the experimental results.

1.—Testing Procedure.

Power Obtainable with Different Fuels at the Same Compression-ratio.

With a view to determine definitely the power output obtainable from each fuel at the same compression-ratio, several hundreds of

tests were run both on the variable-compression engine and on the supercharging engine. In all cases the procedure adopted was as follows :—

A suitable compression-ratio was chosen at which it was known that none of the fuels to be tested would detonate under any conditions of mixture strength or ignition advance. The engine was first started up either on the specially prepared aromatic free petrol or on pure benzene, which was adopted as the standard fuel for high-compression work. It was then run with wide-open throttle at a speed of 1500 revs. per minute until the temperature of the jacket water, carburettor, lubricating oil, and induction pipe had settled down to constant values.

The best ignition setting and mixture strength for the development of maximum power on the fuel were, of course, well known from previous experience, but no readings were recorded unless or until the torque was within $\pm$ 0.3 per cent of that known to be obtainable with the standard fuel (after correction for barometer).

So soon as the running conditions were considered to be in every respect normal, and the engine was found to be developing standard power with standard settings, the fuel was changed over to the sample to be tested ; the correct mixture strength and ignition setting for the development of maximum power with the new fuel were found, and readings of torque were taken. In some cases the torque curve over a wide range of speed, generally from 800 to 1800 revs. per minute, was explored, but this practice was discontinued later after it had been found from repeated experiments that the characteristic of the torque curve over the whole available range of speed was the same for every fuel tested (provided, of course, that the correct mixture strength and ignition setting were used for each speed). The maximum power output obtainable from each sample of fuel was determined in this manner, the time taken to find the best settings and secure steady and reliable readings being about half an hour in each case.

During the whole period of such a series of tests the throttle remained wide open, and great care was taken to prevent any variation in temperature either of the circulating water or the heat-input to the carburettor.

After completing a series of such tests on a number of different samples, a final test was always made on the standard fuel, when, with the same temperatures throughout, and the original adjustments for this fuel, the power output had to return to within less than 0.5 per cent of the initial reading, or the tests were discarded as unreliable. In such tests only one variable remained, namely, the temperature of the lubricating oil, which always rose slightly during a prolonged run of the nature just described. In the case of the variable-compression engine the oil temperature had no appreciable influence on the performance, but in the case of the supercharging

engine the friction losses diminished perceptibly as the oil temperature rose, and a small correction seldom exceeding 0.4 per cent had to be made for this factor. A large stock of lubricating oil was prepared and set aside for the purpose of these tests in order to ensure uniformity throughout.

At the conclusion of each series of tests the engine was motored by means of the balanced electric dynamometer in order to check the mechanical losses.

For the sake of uniformity all figures are given in terms of indicated horse-power, indicated thermal efficiency, etc. This was found desirable because of the differences in mechanical efficiency between the several engines upon which the tests were carried out. The mechanical losses of the engines used were ascertained very carefully by both direct and indirect measurements, and, once determined, any variation could be detected at once by the motoring test. In the case of the variable-compression engine no measurable change occurred after the first thirty or forty hours' running, but in the case of the supercharging engine slight variations in friction were noted from time to time, due probably to slight distortion or lack of rigidity in the structure of the engine.

The power tests at constant compression were run under three different conditions, namely :—

(1) With no heat supplied to the carburettor.

(2) With a constant heat supply of 65 B.Th.Us. per minute when running at 1500 revs. per minute, or 0.0433 B.Th.Us. per revolution when running at speeds other than 1500 revs. per minute.

Note.—In the variable-compression research engine the heat supply to the carburettor is provided by means of a series of resistance coils placed in the air intake passage to the carburettor. The amount of heat supplied to these coils can be measured accurately by means of instruments in the circuit, but it does not follow that the whole of this heat is taken up by the incoming air, though the proportion taken up is a constant one. From the known weight and specific heat of the air it would appear that only about 80 per cent. of the total heat given out from the heater coils is conveyed into the cylinder. In actual running the proportion taken in cannot be determined very accurately, because the thermometer in the induction port acts as a " wet bulb " when unevaporated fuel is entering the cylinder. With a view to checking the proportion of heat taken up by the entering mixture, tests were run with gaseous fuels and also with pure air—the engine, in the latter case, being motored at normal speed, when the quantity of heat taken up by the air entering the cylinder could be determined fairly accurately from its rise in temperature. In this and in future references it should be noted that the " heat-input " refers to the total heat given out by the resistance coils, and not to the actual amount of heat taken up by the air, which, although it cannot be determined with absolute certainty, may nevertheless quite safely be regarded as bearing a constant proportion to the total input, since the weight of air passing over and through the heater coils is constant, as is also their temperature.

(3) With the heat supply varied to maintain the entering gases

at a constant temperature of 15° C. as measured by a thermometer in the induction pipe and close to the inlet valve.

Note.—As mentioned in the preceding note, the figures given for the temperature of the working fluid as it enters the cylinder must be regarded as a first approximation, and are only strictly accurate when the whole of the fuel is evaporated before it reaches the thermometer bulb. No means have yet been found for measuring accurately the temperature of the working fluid when a portion of the fuel is unevaporated. As explained previously, the actual temperature of the working fluid during its entry to the cylinder is relatively unimportant, for it is the total heat input which controls the volumetric efficiency, and therefore the power developed.

In addition, in a few cases the fuel was completely evaporated in a separate boiler and the vapour admitted to the carburettor, thus eliminating entirely any variables due to the latent heat of evaporation.

These tests, however, proved very difficult to carry out, and were confined to a few fuels only. When external evaporation was complete, it was found that the mean effective pressure fell 0.36 per cent per degree C. rise of temperature, which is in tolerable agreement with the theoretical variation in density of the charge.

2.—Experimental Results.

In Table VI., p. 150, are given the mean results of a very large number of tests taken on the variable-compression engine, with the different temperature conditions referred to above. Since actual experimental results are given, the fuels enumerated are of necessity only those of which fairly large quantities could be obtained, being the same as those previously tabulated.

In view of the fact that in most engines the heat supply to the carburettor may be regarded as more or less constant, the results obtained under the conditions shown in Columns B and D are the most important and the most nearly allied to practice. The results shown in Columns C, E, and G are interesting, in that they give an indication of the volatility of the fuels.

In Table II., Columns A, B, and C, show the approximate composition of each of the fuels on which power output was observed. From the composition of each fuel has been calculated (Column A, Table VI.) a figure of merit for power output which is exactly comparable with the values given for relative power output (Column B of Table V., p. 149) deduced for the pure substances in the manner described. It will be seen that it is only for fuels (such as the alcohols) with unusually high latent heat that the observed power output (Columns B or D given in terms of indicated mean effective pressures) falls short of the calculated figures of merit (Column A). Column L, Table II., gives the observed power output in terms of indicated mean pressure for each of the fuels under equal conditions of heating, but after full advantage has been taken of increasing the compression-ratio up to the limit set by detonation or by pre-ignition for that particular fuel.

With the exception of ethyl alcohol and methyl alcohol, which, on account both of their high latent heat and relatively large increase in specific volume, behave somewhat differently, and require special consideration, the following facts may be considered as having been established past all doubt from the Tables given above and from numerous other experiments of a similar nature to those recorded here. Mostly, the facts were arrived at first by purely theoretical reasoning, and they have since been confirmed by actual tests carried out under conditions which ensure a very high degree of accuracy.

3.—Conclusions Arrived At.

(1) That within very narrow limits the power output obtainable from any volatile hydrocarbon fuel, or from any combination of such

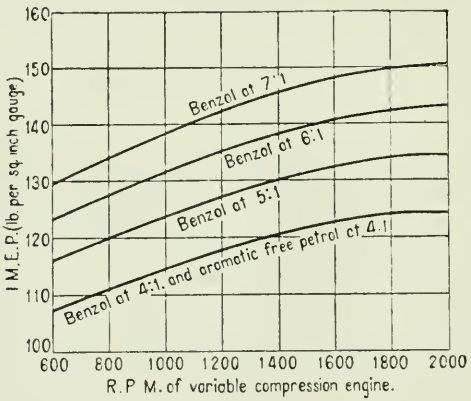


Fig. 15.—Relation between indicated mean effective pressure and revs. per minute in the variable-compression engine at different compression-ratios with benzol and with aromatic free petrol.

Note.—The curve for aromatic free petrol at 4 : 1 compression lies so close to that for benzol at the same compression that it is not possible on this scale to show them separately.

fuels, is substantially the same when used with the same compression-ratio, provided that the compression-ratio is sufficiently low to permit of each fuel being used to its full advantage for that compression.

(2) That so long as the ignition timing is adjusted to suit the normal rate of burning of the fuel, the relative power output is the same at all speeds, *i.e.*, there is no evidence to show that certain fuels, giving more power relatively to another at low speeds, do not also give more power at high speed. This is shown in Fig. 15.

(3) That maximum power is always developed when the mixture strength is at or near that giving complete combustion. (See Fig. 16.)

(4) The power output obtainable from any fuels or combination of fuels, when the external heat supply is constant, or, more strictly, when it is proportional to the latent heat of evaporation of the fuel, is directly proportional to the energy content of the combustible mixture, from which it would appear that the efficiency with which all fuels are burnt is the same. It will be shown later, when dealing

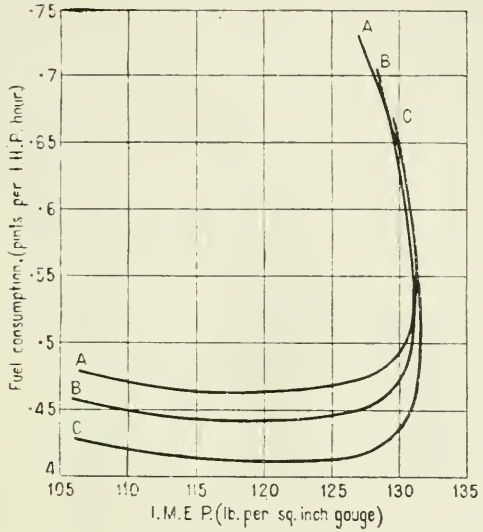


FIG. 16.

Relation between I.M.E.P. and fuel consumption for different fuels at the same compression-ratio, which is sufficiently low to permit of each being used under the most economical conditions for that compression. Curve (A) is with aromatic free petrol, Curve (B) is with "A" petrol. Curve C is with benzol (S.G. 0.880). It will be seen that while the minimum consumptions vary widely, depending (at this low compression) solely on the heat values of the respective fuels, the maximum I.M.E.P., and therefore also the power output, is practically the same for each of these very different fuels.

with efficiency, that within narrow limits this was found to be the case.

In the case of the alcohol group the conclusions may be stated as follows :—

(1) Owing to their low boiling-point it is unnecessary to provide much external heating; consequently the bulk of the fuel enters the cylinder in a liquid state, with the result that the temperature of the charge is lower, and the power output obtainable is considerably greater than from other fuels, although their total energy (see Table III., Column E) is actually somewhat lower.

(2) There is no evidence to show that their low rate of burning has any adverse influence on the power obtainable at very high speeds or very low compression-ratios.

(3) Again owing to the high latent heat and relatively large increase in specific volume, the power output obtainable increases as the mixture is enriched, and continues so to increase until the mixture strength is at least 20 per cent more than that required for complete combustion. (See Fig. 17.)

(4) Owing to the fact that in the case of this group, and this group only, an appreciable proportion of the fuel evaporates during the

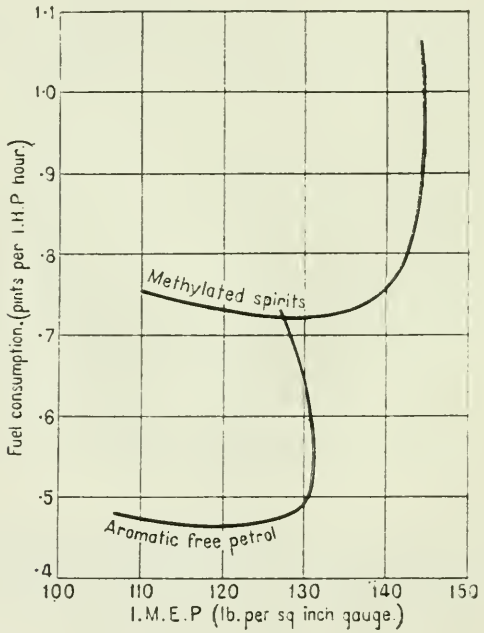


FIG. 17.—Relation between I.M.E.P. and fuel consumption with petrol and with methylated spirits (a mixture of alcohols). The results were observed under exactly equal conditions of compression-ratio, heating and ignition timing. It is seen that while the minimum consumption of the methylated spirits is 55 per cent greater than for the petrol, the maximum I.M.E.P., and therefore the power output on methylated spirits, is 10 per cent greater also.

compression stroke, compression is more nearly isothermal, the temperature of the cycle, as a whole, is lower, and the efficiency is raised thereby. The power output is therefore increased somewhat,

since a slightly greater proportion of the total internal energy available is converted into useful work.

Power Obtainable with Different Fuels and Variable Compression-ratio.

So far only the power output obtainable from different fuels when used with the same compression-ratio has been considered. It is clear, however, that to obtain full advantage of the lower rate of burning of certain fuels each fuel should be run at the highest compression-ratio it will stand without risk of detonation. For the pur-

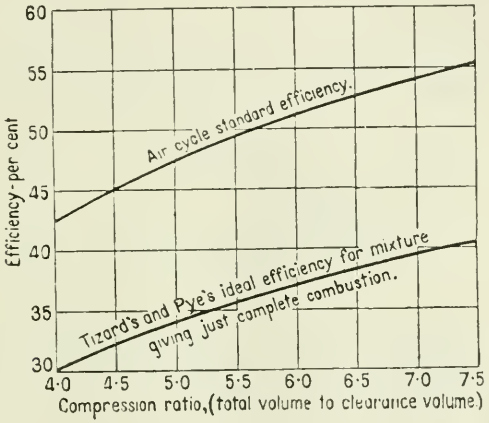


FIG. 18.

pose of these tests the variable-compression engine proved invaluable, for the compression-ratio could be varied to suit each sample within the space of a few seconds, and that without reducing either the speed or the load, or affecting any of the temperature conditions.

Although a very large number of tests was carried out extending over many months, the results obtained may be stated very briefly, for it was found that the relation between power output and compression-ratio was precisely the same in the case of every fuel tested.

1.—EXPERIMENTAL RESULTS.

In all cases it was observed that fuels capable of standing high compressions give, relatively to each other, the same power output at lower compressions. At varying speeds up to the limit at present obtainable the relative power output of different fuels is the same.

At first sight it would naturally be expected that, when the compression is varied without involving any change, either thermal

or mechanical, the mean effective pressure would be proportional at all times to the air standard efficiency corresponding to the compression-ratio used. In Fig. 18 the upper curve shows the air-standard efficiency over the range of compression from 4 : 1 up to 7.5 : 1. The lower curve shows Tizard and Pye's ideal efficiency for any fuel when the mixture strength is that corresponding to complete combustion. This latter curve takes into account the losses due to change in specific heat and to dissociation at high temperatures, but it does not take into account the direct heat-loss to the cylinder walls, which

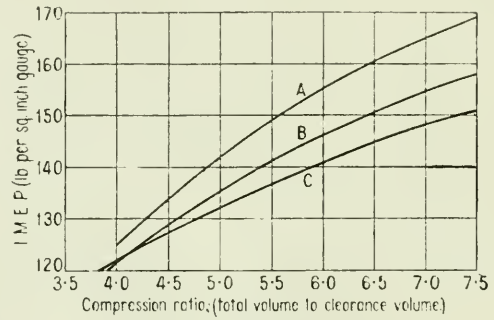


Fig. 19.—Curve (A)—Relation between indicated mean effective pressure and compression-ratio according to Tizard and Pye's ideal thermal efficiency for a mixture strength giving complete combustion, assuming a constant volumetric efficiency of 77 per cent of air at N.T.P. (*i.e.*, 0° C. and 29.92in. of mercury), but without making any allowance for heat-loss to the cylinder head and walls. Curve (B) —The same relation according to the air standard efficiency, assuming a relative efficiency throughout of 65 per cent and a volumetric efficiency of 77 per cent. Curve (C)—The relation of I.M.E.P. and compression-ratio as actually observed.

Note.—It will be shown later that the observed thermal efficiency follows Tizard and Pye's ideal curve very closely. That the observed mean pressure does not do so can only be accounted for by a change in the volumetric efficiency.

is always more or less a variable quantity depending upon the form of the combustion chamber, the speed, and many other variables.

In Fig. 19 curve (B) shows the variation in mean pressure with compression which might be expected if the efficiency relative to the air standard remained constant at all compressions. Curve (A) shows the mean pressure which would be obtained if the efficiency bore a constant relation to Tizard and Pye's ideal efficiency.

Curve (C) shows the relation between the mean pressure and compression-ratio actually observed.

This relation, which could be obtained with a high degree of accuracy, did not vary measurably as between numerous tests on any one fuel or on several different fuels.

2.—Mean Pressure and Efficiency.

That the mean pressure developed does not follow either the air standard or Tizard and Pye's calculated values is explained by the fact that the volumetric efficiency falls as the compression-ratio is increased. Why it should fall still remains a mystery. Many theories have been advanced,* some of which, no doubt, go some way towards explaining the change in volumetric efficiency, but none can, in the author's opinion, explain the existence of so large a variation.

It will be shown later, when dealing with the question of efficiency, that the actual observed efficiency rose in relation to the air standard,

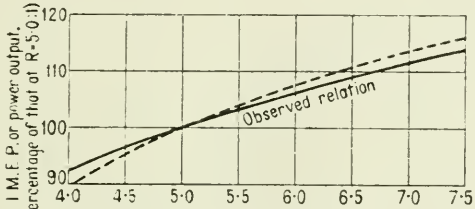


Fig. 20.

The full line shows the observed increase or decrease of maximum I.M.E.P., and therefore of power output when compression is varied, as compared with that obtainable at a compression-ratio of 5 : 1. The dotted line shows what this relation would be if it were proportional to the air standard—that is, if relative thermal efficiency and volumetric efficiency were constant with varying compression. It appears that both are variable, but that with increasing compression the fall in volumetric efficiency more than balances the slight rise in relative thermal efficiency.

while the mean pressure fell in relation to it. This is a somewhat surprising phenomenon, in view of the fact that, so far as any change in the efficiency of the combustion chamber was concerned, it must have decreased with increase of compression, though every endeavour was made to reduce this variable to the lowest possible limit.

Fig. 20 shows in full line the observed percentage increase or decrease in mean effective pressure, taking as basis a compression-ratio of 5 : 1. The dotted line shows what this relation would be if the I.M.E.P. were proportional to the air standard. The full-line curve may be taken as applicable to all fuels or combinations of fuels without distinction or qualification. To apply it, it is necessary

* See Proc. R. Ae. Soc., Nov., 1923, "Thermodynamics of Aero Engines," by Ricardo.

only to refer to Table II., Column D, which gives the highest safe compression-ratio at which the fuel may be used, and to add or subtract the necessary percentage observed. Table II., Column L, shows the mean effective pressure actually observed with each fuel at 1500 revs. per minute when run at the highest compression-ratio consistent with freedom from detonation.

Thermal Efficiency as Affected by Fuel Composition.

Before any serious tests were commenced on the variable-compression research engine, Messrs. Tizard and Pye had undertaken an exhaustive investigation into the process of combustion, and had arrived at very definite conclusions, both as to the chemical changes which might be expected to take place, and as to the limits of thermal efficiency resulting therefrom. An abstract of the results of their investigation formed the subject of a lecture delivered by Mr. H. T. Tizard before the British Association in August, 1920. Subsequent tests on the two research engines, more particularly on the variable-compression engine, and on various fuels, have served to confirm the conclusions they arrived at. The close agreement obtained during tests carried out comparatively recently with their figures deduced many months previously, and from purely theoretical reasoning, is very striking. Since Messrs. Tizard and Pye have dealt so comprehensively with all the factors governing the efficiency of an internal combustion engine, and the influence upon that efficiency of changes in the composition of the fuel, it is unnecessary for the author to give more than the barest summary of their conclusions.

1.—Results Obtained by Messrs. Tizard and Pye.

The thermal efficiency obtainable with any fuel depends in the first place upon the compression-ratio at which it can be burnt, and this in turn is governed by the tendency of the fuel to detonate under high compression-pressures. At any given compression-ratio the thermal efficiency is dependent upon the maximum temperature. Upon flame temperature depends also the direct loss of heat to the cylinder walls, the loss due to dissociation, and that due to increase in the specific heat of the working fluid at high temperatures. The maximum temperature is dependent upon the mixture strength and the available internal energy of the fuel.

Broadly speaking, the results of the theoretical investigations of Messrs. Tizard and Pye have been to indicate :—

(*a*) That when both change in specific heat and dissociation are taken into account the maximum temperature obtained with economical mixture strengths is substantially the same in the case

of all available hydrocarbon fuels, though it is perceptibly lower in the case of alcohol.

(*b*) That the gain in efficiency with increase in compression-ratio is very considerably greater than would be predicted from the air standard formula.

(*c*) That the theoretical thermal efficiency for the most economical mixture strength may be expressed by the formula $E = 1 - \left(\frac{1}{r}\right)^{0.295}$.

This formula takes into account both change of specific heat and dissociation, and represents the ideal efficiency obtainable from a working fluid consisting of air and any volatile hydrocarbon fuel,

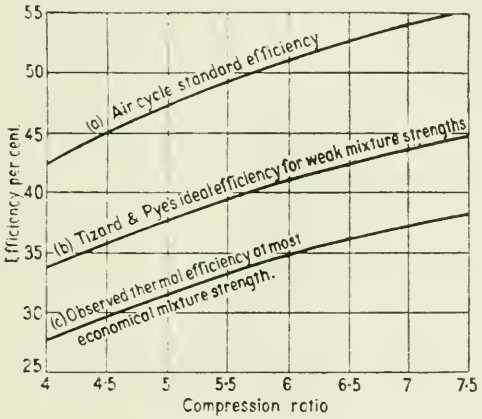


FIG. 21.

assuming complete combustion (within the limits imposed by dissociation) and no loss of heat to the cylinder walls.

In Fig. 21 are shown three curves : (*a*) the air standard efficiency for a range of compression-ratio from 4 : 1 to 7.5 : 1 ; (*b*) Tizard and Pye's ideal thermal efficiency corresponding to the same range of compression ; and (*c*) the observed results on the variable-compression engine with most economical mixture strength based on the net calorific value of the fuel and including the latent heat of evaporation. It will be observed that throughout the whole range the relation between the experimental results and the calculated ideal efficiency is constant to within very narrow limits. The difference between the two represents the heat loss to the cylinder walls during combustion and expansion, and the small loss due to the early opening of the exhaust valves. All the tests showed that the gain in efficiency with increase in compression was almost exactly the same

in the case of all fuels tested. Naturally, the majority of the fuels would not stand so high a compression-ratio as 7.5 : 1 without detonation or pre-ignition, but up to the highest useful compression for each fuel the curves of efficiency plotted against compression-ratio were almost coincident in all cases. Alcohol, for reasons which will be dealt with later, gave a higher thermal efficiency throughout the range of compression, though the curve showed practically the same proportional variation.

2.—The Determination of Calorific Values.

The determination of the true absolute thermal efficiency obtainable from different fuels has proved somewhat difficult. Such a determination presupposes an accurate knowledge of the effective calorific value of the fuel, and it is in acquiring this that the principal difficulty lies. Much of the published data is clearly unreliable and must be discarded, while in many cases the very precision with which the calorific value is given to four or even five places is sufficient to engender suspicion in view of the great difficulty in the way of making an accurate determination. In regard to the application of the calorific values, as determined, to the calculation of engine thermal efficiency, there is a factor which does not appear to have been taken into consideration in previous investigations on this subject. This factor is the influence of the latent heat of evaporation upon the evaluation of the useful calorific value.

When a sample of liquid fuel is tested either in a Junkers or a bomb calorimeter, some at least of the heat liberated during combustion is devoted to overcoming the latent heat of evaporation of the liquid. In the case of the bomb calorimeter, for example, when a sample of liquid fuel is placed in the bomb only a trifling proportion is evaporated before combustion commences. During the process of combustion the remainder of the fuel is evaporated, and the heat required for overcoming the latent heat of evaporation is supplied by combustion. Now when used in an engine the whole, or very nearly the whole, of the fuel is evaporated before the end of the compression stroke, such small proportion as escapes evaporation usually passes through the cylinder unburnt. In practice the proportion of liquid fuel which actually both evaporates and burns during the combustion stroke may be considered as negligible, so that the heat required for evaporation may be regarded as being supplied either outside the cylinder or at least outside the heat cycle. It has been fully accepted by engineers that the latent heat of evaporation of the water formed by the combustion of any hydrocarbon and air should be deducted from the total available heat, because, owing to the high temperature of the cycle, no water is condensed, and consequently this heat cannot be made use of. It is therefore proper and customary to subtract from the total heat the

heat equivalent of the evaporation of the water formed. It appears also to be right and proper to *add* to the total available heat the latent heat of evaporation of the fuel, since this is supplied outside the cycle. To be more strictly accurate there should be added the latent heat at constant volume, which is less than the latent heat at constant pressure by the amount of the work done on the atmosphere if the liquid is vaporised when unenclosed. So far as ordinary petrols are concerned, the latent heat of evaporation of the fuel bears but a small proportion to its total heat, and does not seriously affect the issue, but in the case of the alcohol group its proportion becomes a very considerable one, and should most certainly be taken into account. The most reliable determinations of the calorific value of alcohol, at present available, do not take into account the latent heat of evaporation, and the very high apparent thermal efficiency obtainable with this fuel is therefore largely fictitious.

Theoretical considerations would suggest that the thermal efficiency —that is to say, the proportion of the total effective heat converted into useful work—should be substantially the same at the same compression-ratio for all fuels, with the possible exception of the members of the alcohol group. This is fully borne out by the experiments made. Provided that the fuel is vaporised, or that all liquid is sufficiently finely divided and in suspension, the thermal efficiency at any compression-ratio is practically constant.

In Table VII., p. 151, the following values are given for each fuel :—

Column A.—The lower calorific value according to the accepted standards and the most careful and reliable determinations.

Column B.—The minimum fuel consumption (in terms of lb. per I.H.P. per hour) actually observed with a compression-ratio of 5 : 1 and standard heat input to the carburettor.

Column C.—The thermal efficiency calculated from Columns A and B.

Column D.—The latent heat of evaporation at atmospheric pressure.

Column E.—The "lower" calorific value (Column A) plus the latent heat of evaporation at constant volume.

Column F.—The corrected thermal efficiency calculated from Columns B and E.

For all the data given in this Section, except where specifically mentioned, the thermal efficiency has been calculated from the corrected calorific value of the fuel as given in Column E of Table VII., *i.e.*, the calorific value as determined by bomb calorimeter minus the latent heat of evaporation of the water formed, but plus the latent heat of evaporation of the fuel at constant volume.

In the case of the special research engines, as previously mentioned, the means provided for measuring both the power output and the fuel consumption were such as to ensure an accuracy (based on the mean results) of about 0.25 per cent plus or minus.

It is very doubtful whether the useful calorific value of a composite fuel such as petrol can be ascertained to within closer limits than 0.5 per cent plus or minus, more particularly so when the fuel used is not a chemically pure substance, so that the observed heat value cannot be checked by analysis.

1.—Experimental Results.

Power and Efficiency at Various Mixture Strengths.

The procedure of all tests for power output and efficiency has already been described (page 100 *ante*). Such tests were carried out

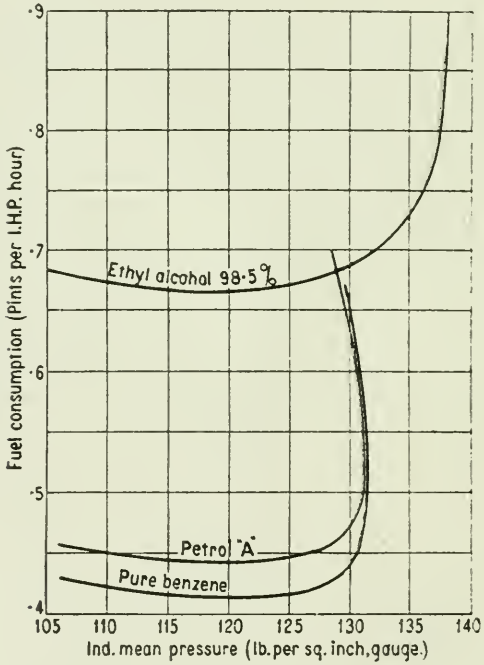


Fig. 22.

on every sample of fuel examined, and when the quantity available permitted, similar tests were run at different speeds, different jacket temperatures, or different rates of heat input to the carburettor. The results obtained from three of such tests are shown in Fig. 22, giving fuel consumption plotted against the mean pressure for petrol "A," pure benzene, and for ethyl-alcohol 98.5 per cent at a com-

pression-ratio of 5 : 1. Fig. 23 shows similar curves for benzene at different compression-ratios, namely, 4, 5, 6, and 7 : 1. It will be seen that the general form and character of the curves so obtained are substantially the same for all the fuels. The differences that do occur are of a small order except in the case of the alcohol group. In the case of alcohol the power continues to rise with increase of mixture strength to an important extent. This is quite certainly due to the increased volumetric efficiency obtained with rich mixtures owing to the high latent heat of the fuel and the consequent fall in the temperature of the charge. The relation between power output

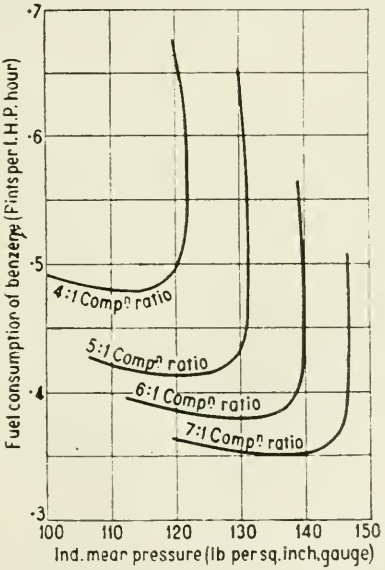


FIG. 23.

and consumption of mixture, on the weak side, is substantially the same for all fuels, but on this side the efficiency obtained depends to a greater extent upon the nature and intensity of the ignition spark and upon its timing. Tests on the weak side were run both with the ignition set to give maximum efficiency at normal mixture, and with the ignition adjusted to suit each reduction in mixture strength. Fig. 24 shows the curves obtained (*a*) with fixed ignition setting, (*b*) with the setting adjusted for each change in mixture strength. It was observed that the ignition required to give maximum power on any petrol with a compression-ratio just below its detonation point and between the limits 4.5 to 5.5 : 1 was 31 degrees before

top centre at 1500 revs. per minute. The ignition could be advanced with advantage when working on the weak side to as much as 45 degrees early on the weakest mixtures. The same applies in the case of all the fuels tested. It is interesting also to note that at about the mixture strength giving complete combustion the ignition timing required was very nearly the same for all fuels from heptane to toluene or alcohol.

While the behaviour of fuels at mixture strengths weaker than that giving complete combustion is very interesting from a scientific point of view, and highly suggestive of possible developments, yet, so far as the multi-cylinder engine is concerned, it is of but little practical value at present, because, owing to irregularities in distribution, such weak mixtures cannot be used. In every instance

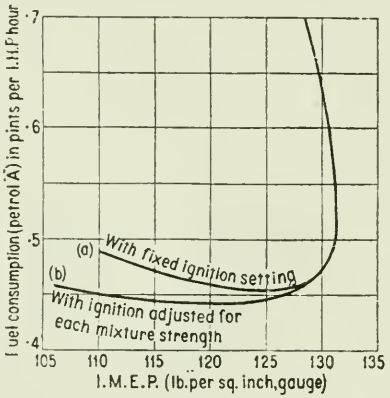


FIG. 24.

where any attempt has been made to supply to a four-cylinder engine a mean mixture strength appreciably weaker than that giving complete combustion, the result has been that one or more cylinders of the group are starved, their ignition becomes irregular, and back-firing into the carburettor takes place.

2.—Effect of Weak Mixtures on Detonation.

It has already been stated, when dealing with the subject of detonation, that the tendency to detonate is at a maximum at or about the mixture strength giving maximum economy. This is practically scientifically true. Strictly speaking, the tendency to detonate is at a maximum at or about the mixture strength giving complete combustion, and this in practice is almost coincident with that giving maximum economy. With the very weak mixtures a rather higher

compression can be used without detonation, but under such a condition an engine is working between exceedingly narrow limits of mixture strength. Satisfactory running would be quite unattainable with an ordinary carburettor even on a single-cylinder engine. In any multi-cylinder engine to work within such limits of mixture strength is certainly far outside practical possibility. With a view to investigating this point more thoroughly a careful series of tests was made on the standard aromatic free petrol over a range of mixture strengths from 75 per cent rich to 25 per cent weak. After each adjustment of mixture strength the compression-ratio was varied so that detonation was just audible. The results of this interesting test are shown in the curves given in Fig. 25. It will be seen that with any mixture strength between 10 per cent rich and

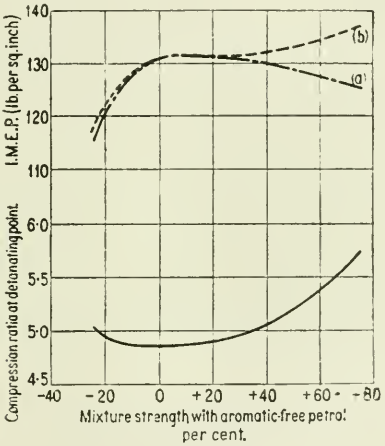


FIG. 25.

10 per cent weak the available compression-ratio was 4.85 : 1. When the mixture was varied in either direction beyond these limits it became possible to raise the compression without causing detonation. The full-line curve shows that when the mixture was 20 per cent weak the compression could be raised to 4.95 : 1, and with a mixture strength 75 per cent rich the compression could be increased to 5.75 : 1. The ignition setting was adjusted to give maximum power for each condition. In the same Figure is shown (*a*) the variation in mean effective pressure over the same range of mixture strength at a constant compression-ratio of 4.85 : 1, and with ignition timing adjusted to suit each mixture strength; and (*b*) the same again, but with compression-ratio adjusted as permitted by each change of mixture strength. From these curves it will be

seen that only a very slight increase in compression-ratio can be obtained with weak mixtures, and that only when the mixture is so weak that the running is becoming unstable.

3.—Influence of Jacket Water Temperature.

While all standard tests were run with a jacket water temperature of 60° C., some series of tests were also run at temperatures of 35° C. and 85° C. In the case of all fuels boiling below about 180° C. the water temperatures did not influence the efficiency to any appreciable extent, though in some cases, notably with the more volatile fuels, a very slight gain in efficiency was observed when the water temperature was at its lowest, due probably to the general reduction in the temperature of the cycle. The difference was, however, exceedingly small, and for all practical purposes may be regarded as insignificant. It must be understood that these remarks refer to the indicated thermal efficiency. So far as the brake thermal efficiency is concerned, any increase in temperature of the cylinder walls results in a decrease in piston friction due to the lower viscosity of the lubricating oil. The gain in mechanical efficiency due to reduced friction losses with higher jacket temperatures generally more than outweighs any difference in indicated thermal efficiency. Since piston friction is purely a mechanical factor and varies widely in different engines, it is thought best to deal only with the indicated performance. In both the variable-compression and the supercharging engines on which these tests were made the piston friction is so low that the brake thermal efficiency is virtually unaffected by changes in the temperature of the cooling water.

4.—Influence of Heat Input to the Carburettor.

In all the standard tests a uniform rate of heat input of 0.0433 B.Th.Us. per minute per revolution per minute, or 65 B.Th.Us. per minute at 1500 revs. per minute, was supplied. Other series were also carried out with no heat input to the carburettor, and also with a heat input of 0.066 B.Th.Us. per minute per revolution per minute.

So far as the more volatile fuels are concerned, that is, fuels boiling below 180° C., the heat input, while affecting the power very adversely by reducing the weight of charge entering the cylinder, had no influence upon the thermal efficiency except in so far as a high rate of heat increased the tendency to detonate, and so necessitated the use of a lower compression-ratio. The effect of heat input to the carburettor and of water jacket temperature when running with a mixed fuel having an abnormally wide range of boiling, will be dealt with later, but the above remarks must be regarded as applicable to reasonably volatile and homogeneous fuels such as the various

petrols given in the Tables, also the lighter aromatics and naphthenes. In the case of 95 volume per cent ethyl alcohol the efficiency rose from 31.6 per cent to 32.5 per cent, when the heat input was raised from nil to 0.066 B.Th.Us. per minute per revolution per minute. In view of the behaviour of most of the petrols which have very much higher final boiling points, this feature is perhaps a little curious ; but alcohol, owing to its higher latent heat, its considerable water content, and its low vapour tension, behaves somewhat differently from most of the other fuels tested. Fig. 26 shows the mean effective pressure and fuel consumption obtained with 95 volume per cent ethyl alcohol, curve (*a*) with no heat input, and curve (*b*) with a heat input of 0.066 B.Th.Us. per minute per revolution per minute. It will be seen from these curves that the minimum fuel consumption is appreciably lower under the heated condition. The maximum power output, of course, is very much lower. From tests on multiple-cylinder engines on petrol, benzole, etc., where the problem of distribution enters into the account, it was found that a very considerable heat input was required in order to obtain the best therma

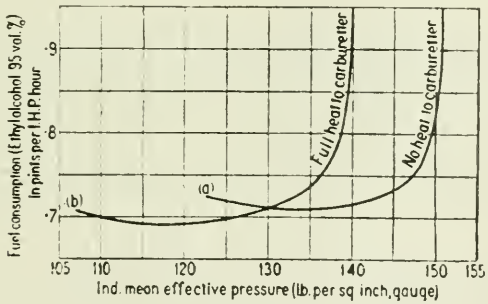


Fig. 26.

efficiency. The heat required varied considerably for different engines ; the amount may be taken as an inverse measure of the efficiency or otherwise of the distribution system.

5.—Efficiency on Reduced Loads.

When the load on an engine is reduced by throttling the supply of working fluid, and the mixture strength is maintained at the value used for full load, the temperatures in the heat cycle will be substantially the same, since neither the ratio of compression nor the heat of combustion of the mixture are altered. The only condition which is changed appreciably is the proportion which the residual exhaust products bears to the whole of the working fluid. Apart

from the latter consideration the efficiency of the cycle should remain very nearly the same at all throttle openings. Experiments with the time of ignition fixed at that giving maximum efficiency on full throttle have shown that, as the throttle is closed and the power output is reduced at constant speed and constant mixture strength, the efficiency falls; but when the ignition is readjusted to give maximum efficiency at each throttle opening, the indicated thermal

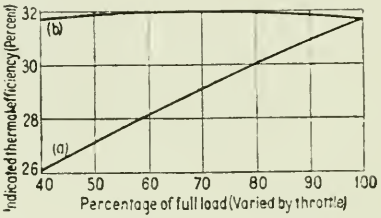


FIG. 27.

efficiency remains substantially constant over a wide range of load, *e.g.*, from 40 per cent to 100 per cent. Fig. 27 shows (*a*) the indicated thermal efficiency observed on petrol with a constant mixture strength and fixed ignition setting, and (*b*) the indicated thermal efficiency with the ignition adjusted at each load to give maximum efficiency. It will be observed in the case of curve (*b*) that the efficiency at about 70 per cent full load is actually appreciably greater

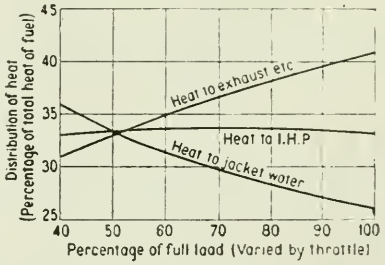


FIG. 28.—Petrol "A." Compression 5.45 : 1.

than that obtainable on full loads. This characteristic is not peculiar to petrol, but has been observed also with alcohol. Very great care was taken to ascertain carefully the variation in pumping losses at the different throttle openings, and to correct for such variation when computing the indicated power or efficiency. It would appear that the presence of a greater proportion of exhaust products on

reduced loads influences appreciably the rate of burning of the mixture, and therefore necessitates an earlier ignition timing.

Fig. 28 shows the variation in heat distribution for petrol "A" with different loads at a constant speed of 1500 revs. per minute and under uniform conditions as to heat input, jacket temperature, etc.

Fig. 29 shows a similar set of curves for 95 volume per cent alcohol under identical conditions. In both cases the heat input to the

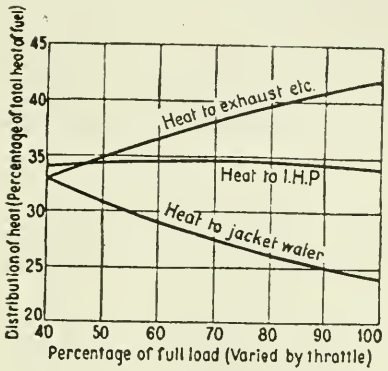


FIG. 29.—Alcohol 95 vol. per cent. Compression 5.45 : 1.

carburettor was proportional to the load and equivalent to 65 B.Th.Us. per minute at full load.

6.—EFFICIENCY ON ALCOHOL.

While at any given compression-ratio the thermal efficiency was found to be the same on all volatile hydrocarbon fuels, the efficiency obtained with alcohol was consistently higher. For the samples of ethyl alcohol and for a sample of methylic alcohol (purified wood naphtha) accurate calorific values were obtained. These determinations were made by Messrs. The Asiatic Petroleum Co. at their central testing laboratory at Shell Haven. The determination made of the heat value of 98.5 per cent ethyl alcohol agreed with the value given by Richards to within 0.25 per cent, which, considering the difficulties which are encountered when making these determinations, is very close agreement indeed. The values given by Richards are recognised at the present time as being probably the most accurate in existence. The calorific values, as determined at Shell Haven, of the actual samples used for the engine tests may be taken, therefore, as being as nearly as possible accurate. The values for the alcohol are given in Columns A and E of Table VII., p. 151. Very careful tests were run with 95 volume per cent ethyl alcohol at

compression-ratios of 3.8 : 1, 5 : 1, and 7 : 1 under exactly similar conditions as to speed, jacket temperature and rate of heat input to the carburettor, also the distribution of heat to the jackets and to the exhaust was very carefully determined. Similar tests were run also on several petrols (within the limits of useful compression-ratio) and on benzene. The results obtained are shown in Table VIII., p. 154.

From this Table it will be observed that the thermal efficiency obtained with ethyl alcohol at all compression-ratios was some $2\frac{1}{2}$ per cent higher than that obtainable with hydrocarbon fuels, when the calorific values in all cases are subjected to the same corrections for the latent heat of evaporation. The explanation of this gain in efficiency is to be found in the reduction in the temperature of the cycle due to the evaporation of the alcohol within the cylinder.

7.—Heat Distribution.

It has already been shown that the drop in temperature of the working fluid due to the evaporation of the liquid fuel was, in the

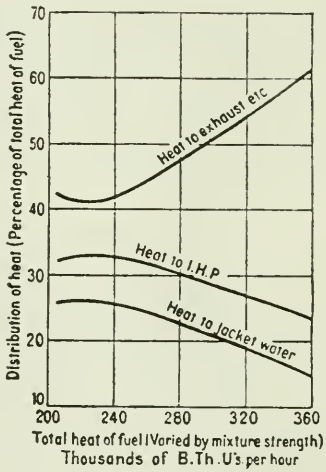


Fig. 30.—Petrol "A." Compression 5.45 : 1.

case of petrol "A," 20 degrees C., and in the case of ethyl alcohol 98 per cent, 85 degrees C., while in the case of 95 per cent volume it would, owing to the high water content, be equivalent to 97 degrees C. There appears to be no doubt that practically the whole of the fuel and the water contained in it evaporates completely either during entry to the cylinder, or, at any rate, before the completion of the

compression stroke; it is clear, therefore, that the temperature at the end of compression will, in the case of alcohol, be some 70 degrees C. lower than in the case of petrol. Since the maximum flame temperature of an alcohol mixture is also about 100 degrees C. less than that of a petrol mixture, it follows that in the former case the highest flame temperature attained will be approximately 170 degrees C. lower than that of a petrol mixture, and the loss of heat to the jackets and that due to change of specific heat, etc., will be substantially reduced. Again, when running with alcohol, the total weight of working fluid passing through the cylinder is greater as explained previously; hence the relative loss of heat will be even slightly less than would appear from a consideration of the temperatures alone.

Table IX., p. 154, shows the heat distribution observed when running with petrol "A" and with 95 volume per cent ethyl alcohol with

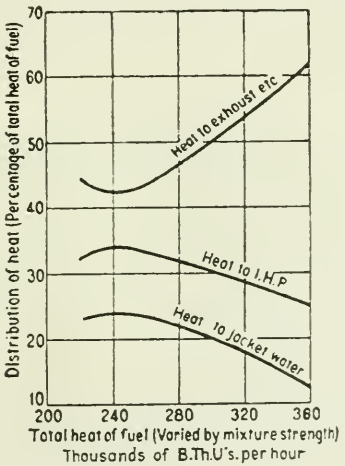


FIG. 31.—Alcohol 95 vol. per cent. Compression 5.45 : .

a compression-ratio, in this instance, of 3.8 : 1. The mixture strength both for the petrol and for the alcohol was that giving maximum efficiency. From a comparison of the proportionate heat loss to the cylinder cooling water, it will be observed that the amount of heat given up to the cylinder walls is considerably greater in the case of petrol than alcohol, indicating again a considerably higher cycle temperature.

Figs. 30 and 31 show respectively the observed variation in the distribution of heat (to indicated work, to cooling water, and to exhaust) with different mixture strengths on petrol and alcohol in both cases at a compression-ratio of 5.45 : 1. The horizontal scale

denotes the total number of B.Th.Us. per hour supplied to the engine, and the vertical scale the percentage accounted for in I.H.P. and cooling water. The upper curve is plotted by difference, and represents the heat rejected in the exhaust and by radiation.

Table XII., p. 152, gives the observed minimum fuel consumptions in lb. and in pints per indicated horse-power per hour. Column A shows the consumptions for a fixed compression-ratio of 5 : 1. Column B gives the minimum fuel consumption obtained when running the engine with the highest useful compression-ratio for each fuel.

STARTING, DISTRIBUTION, AND VOLUMETRIC EFFICIENCY.

It is now proposed briefly to examine the conditions governing ease of starting, distribution, and volumetric efficiency, and to consider in what ways these conditions react favourably or otherwise towards one another.

1.—Starting.

Ease of starting must be considered from two different aspects, namely, (1) the ease with which an engine may be induced to start running under its own power, and (2) the time taken to warm up sufficiently to give full power.

It is common experience that while a motor-car engine may be started up with ease it may take as much as five or six miles running before the engine will "pull" well, more particularly when the carburettor setting is such as to deliver a rather weak and, therefore, an economical mixture.

The conditions governing the ease or otherwise of gaining the initial start will be considered first. They have proved extraordinarily difficult to determine because so many factors enter into the case, such as the condition of the sparking-plugs and the strength of the spark, which has usually to be provided from a magneto running at a very low speed, and also the speed at which the engine is rotated when starting.

To start an engine from cold it is necessary to have present in the cylinder at the time of ignition a sufficient proportion of vapour to form an explosive mixture. The formation of this vapour must be effected notwithstanding the fact that not only the whole of the induction system, but the cylinder walls, pistons, and valves are all quite cold, and also that the velocity in the induction system is at a minimum.

If, at starting, the carburettor were supplied only with a mixture of strength just sufficient to give economical running under normal conditions, the engine would certainly never start from cold with even the most volatile petrol. On the other hand, if an excessive

quantity of liquid fuel be admitted to the cylinder, or spread over the internal walls of the induction system, the large surface of liquid thus exposed will allow a sufficient quantity of the more volatile fractions, even of the least volatile of the present-day petrols, to evaporate and so produce a combustible mixture. Even in cold weather, therefore, it is possible to start an engine from cold with any reasonable fuel.

From these considerations the following conclusions may be drawn :

(*a*) On no petrol will an engine start from cold with an economical mixture strength.

(*b*) On commercial petrol an engine will start from cold provided sufficient fuel of reasonable quality is admitted.

In all fuels belonging to the group known as petrol, some of the light fractions, which occur in the majority of crude petroleums, exist. These light fractions consist either of hexane, cyclohexane or benzene, and often of all three of these substances, and their presence renders starting from cold possible.

In all modern carburettors some means, such as a pilot jet, is provided in order to deliver a very rich mixture for starting and slow running, and this is generally supplemented by " flooding " when starting from cold. How much " flooding " is required depends primarily upon the arrangement of the induction system. When the induction pipe is of large diameter so that when turning by hand the velocity through it is very low, and when the carburettor is fitted at the lowest point with efficient drain holes, it is sometimes found that no amount of " flooding " will suffice for the less volatile petrols, and that fuel must be injected either into the cylinders or into the upper portion of the induction system. On the other hand when the induction system is of small cross section so that even when turning by hand the velocity is high enough to raise and spread the liquid fuel over the whole surface of the combustion chamber, an engine will usually start readily enough after " flooding " the carburettor even with a petrol of low volatility. If the mixture from the pilot jet is sufficiently rich it will often start the engine without any " flooding " at all. In some cases the induction system is so arranged that when the carburettor is " flooded " freely the quantity of liquid fuel taken into the induction system and cylinders is so large that the proportion of vapour to air becomes too great, and the engine will fail to start from over-richness. The ease, therefore, with which an engine may be started from cold may be said to depend upon the following :—

(*a*) The arrangement and proportions of the induction system and carburettor combined with the degree of " flooding " which is possible.

(*b*) The adequacy of the pilot jet to supply a comparatively large quantity of very rich mixture, and

(*c*) The proportion of hexane, cyclohexane or benzene or other

very volatile fraction in the fuel which determines the amount of "flooding" or "doping" required.

With the least volatile of the petrols tested it was always found possible to provide an over-rich mixture on the coldest days, either by "doping" the cylinders or, in some cases, merely by excessive "flooding" of the carburettor, depending on the induction system design.

The conclusions so far recorded apply to the composite fuels known as petrols. The statements are not applicable to pure substances having constant boiling points. For instance, it was found almost impossible to start from cold on pure heptane, even with the most careful "doping" of the cylinder and any degree of "flooding." This test is interesting since the boiling point of heptane is practically the same as the average of that of petrol "F," which is a very light aircraft spirit. Table X., p. 154, shows a comparison of these two fuels.

The inlet temperature as shown in column C of this Table gives (as will be described later) an approximate measure of the mean volatility.

Column D shows the vapour pressures at 0° C. of these fuels. It will be seen that while the mean boiling point and mean volatility (as indicated by induction-pipe temperature) are practically the same for these two fuels, the vapour pressure under the same low temperature is very different. With the fuel with the high vapour pressure the engine started from cold with extreme readiness, while with heptane it would not start at all.

It is evident that in so far as the influence of the fuel is concerned the vapour pressure is indicative of the readiness of a fuel to start from cold. As will be shown later, the mean volitility, on the other hand, largely controls the time taken to get fully under way.

There are other considerations bearing on the problem of starting from cold when fuels other than petrol and benzol are used. Two of these new factors are :—

(*a*) The ratio of fuel to air required to provide an inflammable mixture, and

(*b*) The heat required to supply the latent heat of evaporation.

For any petrol and even for a commercial benzol, both the fuel to air ratio (by weight) and also the latent heat of evaporation are approximately the same. Up till now, therefore, these variables have been neglected in regard to their effect upon ease of starting. So soon, however, as the alcohols and allied substances come under consideration as fuels these new factors must be taken into account. Table XI., p. 154, shows a comparison of some typical pure substances. The first three, namely, hexane, cyclohexane, and benzene represent the volatile portions of most petrols.

From this Table (Column D, which gives the ratio between the

weights of air and fuel required to provide a mixture giving just complete combustion) it is seen that for the volatile constituents of petrol approximately one-fourteenth of the weight of air is required. For ether one-eleventh of the weight of the air gives a correspondingly inflammable mixture, and for ethyl alcohol as much as one-ninth of the weight of the air present must consist of fuel vapour. From Column C of Table XI. it is seen that there is comparatively little variation between the latent heats of the fuels named except for alcohol. The latent heat absorbed in vaporising a pound of alcohol is no less than $2\frac{1}{2}$ times as much as that required to vaporise the same weight of petrol. Approximately, 1.6 times the weight of alcohol is required to give an inflammable mixture with a given cylinder full of air, so that almost exactly four times the quantity of heat is required to furnish a combustible mixture with alcohol as compared with petrol.

As a practical means of using alcohol the addition of comparatively small proportions of ether, with its high vapour pressure, renders starting quite easy.

2.—Initial Running Time Taken to Attain Full Power.

So far, the problem of effecting the initial start only has been considered. As to how long a period must elapse before the engine will develop its full power depends on quite other factors than those already mentioned. The most important probably are :—

(*a*) The mean volatility of the fuel.

(*b*) The arrangements provided for warming the carburettor or induction system.

(*c*) The design of the carburettor, and

(*d*) The equality of distribution of the fuel to the different cylinders.

The term "volatility" here may be taken in its broadest sense, meaning the readiness with which a liquid vaporises under any given temperature conditions. Many tests were carried out on the road, with different petrols, for the purpose of investigating roughly the effect of volatility upon the time taken in getting under way with a motor car. In each case the engine was started up from "all cold" and immediately driven along a level road with the carburettor set to give rather a weak mixture. The distance travelled before top gear could be taken was carefully noted, and the results were found to agree generally with the volatility tests, which will be described later.

Much clearly depends upon the arrangements for heating the induction system. When the passages are water jacketed the heating up is naturally very slow because of the large heat capacity of the system ; on the other hand, when the system is exhaust heated, the heat capacity is relatively small and it rapidly attains its normal working temperature.

Carburettor design naturally influences greatly the readiness with which power and flexibility of running are gained after starting a cold engine. If the means of compensation are effective, thereby eliminating "blind spots," *i.e.*, conditions of throttle opening or engine speeds where the mixture becomes unduly weak, then the sooner will it become possible to get the engine under full power, even when it is not yet thoroughly warmed up.

Equality of distribution is hardly less important than adequate compensation in the carburettor in its effect upon the time taken to get under way. If we assume that the carburettor delivers to the induction system a mixture of normal strength, then so long as the distribution is uniform there is still some range available on the weak side. Consequently, regular running will be obtained before the engine and induction system have reached their normal temperature. With poor distribution, whereby one or more cylinders receive a mixture weaker than the rest, the available range of mixture strength is much reduced, and consequently until everything is well warmed any attempt to open the throttle is frustrated by "popping back" due to the after-burning caused by excessive weakness of mixture in one or two cylinders.

3.—Mean Volatility of the Fuel.

In the preceding notes on starting use has been made of the term "mean volatility." Before dealing with the subject of distribution it will be well to define exactly what is meant by mean volatility, which, so far as the nature of the fuel is concerned, is the most important factor controlling the time taken to obtain full engine power.

The method used for measuring the heat supplied to the entering mixture on the experimental engine has already been described. By "motoring" the engine at definite speeds, by means of the electrical dynamometer, it was found possible to determine with accuracy the temperature which would, under running conditions, be recorded in the induction passage for any given supply of heat to the air if no fuel were supplied to the carburettor. When the fuel was turned on and its flow was adjusted to give a suitable standard mixture strength, the fall in temperature, caused by the latent heat of evaporation, was noted for different fuels. Having ascertained the constants for a number of the standard fuels by means of tests made with the engine "motoring," it was found that the behaviour as regards vaporisation of the various fuels could be studied with equal accuracy when the engine was actually running under its own power.

By means of motoring tests while only air was admitted to the carburettor it was possible to determine the proportion of the heat which was lost, presumably, mainly owing to the slight blow-back which occurs at inlet valve closing. This test, which was made at

various engine speeds, consisted in rotating the engine while cold (and without fuel) at a constant speed, and supplying the air-intake heater with a known quantity of heat, the difference between the temperature of the air in the induction pipe and that of the outside air being noted.

Since both the specific heat and the weight of air passing were known within fairly close limits, the actual quantity of heat taken into the engine could be calculated. In this determination the only assumption made was in regard to rate of flow of air. Subsequent tests, some of which are described below, made with a displacement air meter, showed that the estimated flow, taken from calculated volumetric efficiency, agreed with the measured flow within such

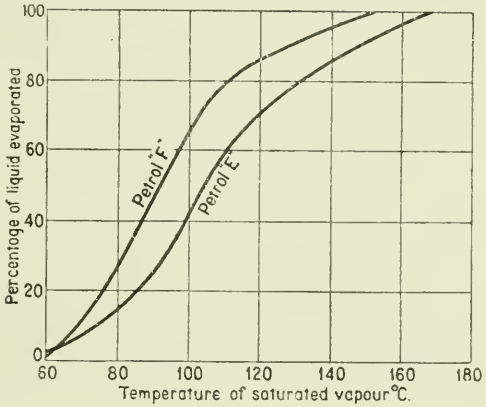


FIG. 32.—Curves showing relation between temperature of the saturated vapour and the percentage of the total fuel boiled off when each temperature is reached, obtained by distillation in an Engler flask.

close limits as to affect the results of the heater tests to only a negligible degree.

Having determined the corrections required for measuring the actual heat supply under any conditions of engine speed, tests to measure the effective volatility of the fuel under running conditions could be made.

The standard test made on each of the fuels referred to in these articles included the following readings :—

(*a*) Temperature recorded by thermometer in the induction pipe close to the inlet-valve housing.

(*b*) Temperature of the outside air before entering the heater.

(*c*) The net heat input to the carburettor.

(*d*) Fuel consumption.

For the standard test the net heat input was maintained constant at 0.0433 B.Th.Us. per revolution of the engine, as before. The fuel consumption was adjusted as nearly as possible to that giving a mixture strength 20 per cent greater than that required for complete combustion, this being the mixture strength at which maximum power is obtained with fuels such as petrol and benzol. The temperature rise or fall, *i.e.*, the difference between reading (*a*) and reading (*b*), is given in Table XII. (p. 152) for most of the fuels tested. (See Column E.).

In Table I. (p. 142) is shown (Column G) the boiling point or boiling range of each of the fuels named. In Column C the latent heat per lb. is given, and in Column A the vapour pressure at 0° C. is indicated. Finally, in Column F, Table III., the air-fuel ratio for just complete combustion is shown.

An examination of the Tables shows that the temperature rise above atmospheric temperature (or the fall below that temperature in the case of the very low boiling point fuels) varies very consistently with the boiling point or with the average boiling point as given by the Engler distillation curve. It may be seen, also, that the temperature rise bears no relation whatever to the vapour pressure. For example, in Fig. 32 the distillation curves of petrols "E" and "F" have been drawn from the values shown in Tables I. and III. While, as will be seen from the Tables, these two fuels have almost exactly the same vapour pressure, the average boiling points as shown by the curves are approximately 104° C. and 92° C. respectively. The temperature as measured in the induction pipe indicates the difference in the mean volatility very clearly. For petrol "E" the temperature rise was 10.5 degrees C., while for petrol "F" it was only 5.5 degrees C.

Since the lower temperature must be caused by more rapid evaporation within the induction system, these temperatures may fairly be taken as a good indication, relatively of the effective or mean volatility of the fuel.

Temperature readings are given in Table I., Col. J, for alcohol and other substances having high latent heats. In these cases, it must not be assumed that the measured temperature is always a true indication of the rate of evaporation taking place within the induction pipe, since, where there exist considerable differences in latent heat, an exact comparison becomes altogether too complicated.

4.—Distribution.

To obtain equality of mixture strength or good distribution to a number of cylinders fed by one carburettor is one of the most difficult and complex problems which confronts the designer of an internal combustion engine. No attempt will be made to deal with all the mechanical considerations which have to be taken into account,

since anything approaching a complete treatise on this subject would be far too lengthy to include here ; moreover, a complete knowledge is by no means available. In passing, however, it may be remarked that for three-, four-, six- and eight-cylinder engines complete mechanical symmetry is impossible if only one carburettor is used.

At a time when so much still remains to be learnt as to the conditions governing distribution in a multi-cylinder engine it would be unwise to attempt to draw too many conclusions as to the influence of the nature of the fuel upon distribution. This much, however, seems to be fairly clear, namely, that the uniformity of distribution, that is, the uniformity of mixture strength supplied to each individual cylinder of any group drawing from one source of supply, depends primarily upon the design of the induction system generally and to a surprisingly small extent upon the nature of the fuel. A really well designed and efficient distribution system appears to maintain nearly the same uniformity of mixture strength on all fuels, while a badly designed system will give poor results on all fuels, though its defects are accentuated by a fuel of low volatility.

It is concluded from observations on various multi-cylinder engines that the question of distribution and its relation to the fuel must be considered from two independent standpoints :—

(1) The uniformity of the mixture strength supplied to each individual cylinder from any one source of supply.

(2) The uniformity of the mixture strength delivered to the induction system as a whole.

With regard to the first of these points, it may safely be accepted from the outset that the mixture entering the cylinder under almost all conditions of running consists of a non-homogeneous mixture of air, fuel vapour and liquid fuel. Only when running throttled, and then probably only after the induction system has been warmed by previous running, can it be assumed that the charge reaches the inlet valves as a mixture of vapours only.

It is clear that since both the liquid fuel and the air are travelling at approximately the same velocity the inertia of the former is greater, and that to avoid inequality of supply as between different cylinders it is most important that the induction system shall be as symmetrical as possible. Provided that not more than three cylinders are drawing from any one source of supply, it is not difficult to design a reasonably symmetrical and uniform induction system, but so soon as the number drawing from one carburettor exceeds three the difficulty is greatly increased, for not only is the sequence of the induction strokes disturbed, but the effective induction periods themselves overlap.

From the above considerations it will be apparent that the uniformity of distribution as between individual cylinders is mainly a problem of induction-pipe design, and only in a secondary degree is it affected by the nature of the fuel. The extent to which the nature

of the fuel influences equality of distribution is most difficult to investigate accurately. No suitable apparatus as yet exists on which the behaviour of fuels in this respect can be fully explored.

5.—Influence of Fuel on Uniformity of Mixture.

The next point to be considered is the influence of the nature of the fuel upon the uniformity of the mixture delivered to the induction system as a whole. To appreciate this problem we must try to visualise the changes that occur in the induction system when the load is varied. What occurs in any manifold appears to be substantially as follows :—

When the engine is throttled down and running light the absolute pressure in the induction system may be as low as 5 lb. per sq. in., or about 10 lb. per sq. in. below atmospheric pressure. The effect upon the liquid fuel is exactly as if its mean volatility had been suddenly increased. Its latent heat is practically unaltered, but it will tend to evaporate at a much lower temperature. The latent heat required to produce vaporisation is obtained partly by lowering the temperature of the small quantity of air that is drawn in under throttled conditions, but also, and probably chiefly, by conduction through the carburettor and induction pipe. If now the throttle is opened, the pressure within the induction system rises immediately, and a large proportion of the entering fuel is precipitated on the previously dry walls. Such precipitation then continues until the walls are thoroughly wetted, the thickness of the layer of liquid depending upon the design of the system generally, also upon the temperature of the walls and the velocity of the gases. For any given design of system the lower the velocity, and therefore, also, the lower the engine speed, the greater the thickness of the layer so formed. Now unless the carburettor supplies a very rich mixture at this stage the cylinders will receive only a weak mixture until the requisite layer has been formed, with the result that the engine will either stop altogether or will fire back into the carburettor due to the slow burning. If now the carburettor be so adjusted as to supply under these conditions a mixture sufficiently rich in fuel, not only to meet the condition of supplying liquid to line the walls, but, at the same time, to supply over and above this demand sufficient fuel to form a combustible mixture in the cylinders, it follows that so soon as the layer has been formed the mixture supplied thereafter will be excessively rich. In order to obviate the difficulty, many designers endeavour to raise the temperature of the induction system so as to evaporate at once any fuel which may be precipitated thereon.

In order to accomplish this result with most fuels of low mean volatility, it will be necessary to maintain the temperature of the walls unnecessarily high for ordinary running. This clearly will have the effect of decreasing the available power and of increasing the

tendency to detonation at a moment when detonation is most liable to occur.

It would appear to be far preferable to accept the necessity for the formation and existence of a layer of liquid in the induction system and to cater for its immediate formation by so designing the carburettor that it will, immediately the throttle is opened, supply a good gulp of liquid fuel, and thereafter continue to supply mixture of normal strength. In such carburettors as the Zenith or the Claudel-Hobson means are provided for supplying a sudden gulp of liquid, which has collected in the diffuser or compensating-jet chamber, but although these carburettors are designed to do this, the quantity stored up and delivered suddenly is generally quite insufficient to meet the needs of a cool induction system and fuel of of low volatility. If the capacity of the storage well or diffuser tube could be made adjustable it would probably be found possible to obtain instant acceleration with almost any petrol and without the necessity for adding more heat than is needed to prevent freezing in cold weather. This conclusion is made on the assumption that the system is such that equal distribution of mixture between the cylinders is derived without the addition of heat. At present, it is seldom that a design is met with in which it is not necessary to add considerable heat to the system in order to mitigate the consequences of what are, in fact, defects in design. It would appear that the efficiency of an induction system may be gauged by the amount of heat required.

The actual experiments carried out with different fuels all tended to confirm the conclusions arrived at. Tests were made on a number of different multiple-cylinder engines with good, bad and indifferent induction systems. It was confirmed that those engines with bad distribution systems gave poor results on all fuels, though their performance was improved by the use of the more volatile fuels. On the other hand, engines having efficient induction systems were affected by changes of fuel volatility to a very small degree.

As already explained, most of the multi-cylinder engines employed for this research proved so inefficient and so sensitive to large variations in friction, etc., that truly comparative results were seldom obtainable, while the more efficient research engines having each only one cylinder naturally could not be utilised on the problems of distribution.

While many experiments have been made and a great many data collected as to the variations in power, output and efficiency with various fuels when used in multiple-cylinder engines, the author feels that none of them are sufficiently reliable to justify publication. To obtain reliable information on this subject it would appear to be necessary to design and build a special engine equipped with several different types of induction system and all the features necessary to ensure consistency of running, which is the first essential condition

of any reliable scientific tests. Whether, in reference to motor fuels, the information obtained would be of sufficient value to warrant the heavy expense of such an undertaking is, in the author's opinion, open to question. Such information, however, would be of very great value to the designer and manufacturer of engines, for it seems clear that the problems relating to distribution are problems of design in the first place, and that the nature of the fuel plays but a secondary part. As a piece of engine research it is, in the author's opinion, by far the most urgent and important of any which could be undertaken at the present day. It is the one factor about which least at present is known, while it exerts a more powerful influence upon the performance of an engine than almost any other.

6.—Volumetric Efficiency.

Measurement of Air Supplied.

To commence with, it will be well to define what is meant here by volumetric efficiency. Volumetric efficiency is defined as the ratio of the weight of air drawn into the cylinder per cycle to the weight of air at 0° C. temperature and at 29.92in. of mercury pressure which would just fill the volume swept by the piston in one stroke.

During the latter period of the fuel research means have been available for measuring accurately the weight of air taken into the engine. Attempts to work with calibrated orifices proved very disappointing. They yielded somewhat erratic and wholly unconvincing results, also the very serious difficulty of calibrating the orifices and their "sensitiveness" to the sharpness of the edge rendered this method of measurement unreliable. In order to obtain a correct value for the mixture strength used, and a correct determination of the efficiency with which the fuel was burnt, it was essential to provide means for measuring the weight of air taken into the engine not only in relative, but also in absolute terms.

After much consideration it was decided to construct a gas holder having sufficient capacity to supply all the air required to consume the measured quantity of fuel at the weakest possible mixture strength. The details of this arrangement will be dealt with later, see pp. 211–213, and for the present it is probably sufficient to state that means are now provided whereby both the number of cubic feet of air and the number of revolutions of the engine during the consumption of a measured quantity of fuel, are recorded automatically and simultaneously, the former to within ± one-fifth of a cubic foot, and the latter to within ± 4 revolutions. The air from the gas holder passes on its way to the carburettor through an expansion chamber of 8 cu. ft. capacity, which is interposed in order to damp out oscillations in the pipe. The temperature as it passes through this chamber is measured by means of a thermometer. In all cases, the measure-

ments recorded here of air consumption and of volumetric efficiency are reduced to terms of standard temperature and pressure, *i.e.*, 0° C. and 29.92in. of mercury.

So far the apparatus has been in operation for a short time only,* but the following tests have been carried out. The tests on volumetric efficiency referred to below were carried out with a different combustion head from that used throughout all the preceding tests, this new head being the one employed for the tests on alcohol carried

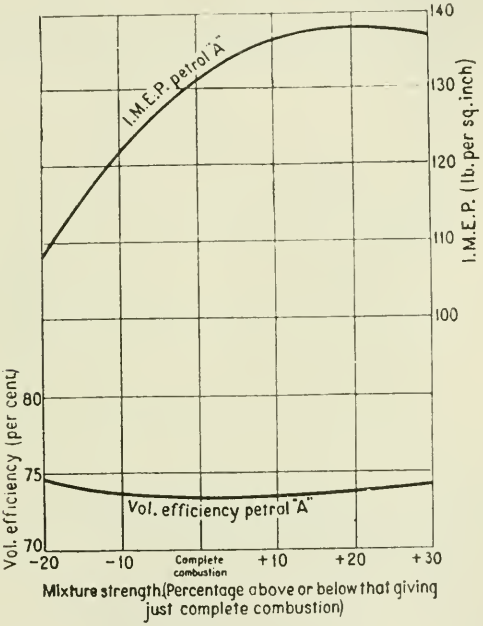


FIG. 33.—Curves showing variations of volumetric efficiency and I.M.E.P. with varying mixture strength. (Compression-ratio constant at 5 : 1 and all other conditions constant.)

out for the Empire Motor Fuels Committee. With this combustion head both the mean effective pressure and the thermal efficiency of the engine are substantially higher than with the head used previously. It will be observed, for instance, that in Figs. 33, 34, and 35 the mean effective pressure under any given set of conditions is much higher than that recorded hitherto; this difference, however, does

* Later work with this apparatus was described in a paper given before the S.A.E., entitled "Some Recent Research Work on the Internal Combustion Engine." See *Automobile Engineer*, Sep., Oct., and Nov., 1922.

not affect in any way the deductions to be drawn from the volumetric efficiency tests, and the author mentions it only to explain what might otherwise appear to be a glaring discrepancy. The tests which have been made are as follows :—

7.—Experiments on Volumetric Efficiency.

(1) Tests of volumetric efficiency at constant mixture strength with constant heating and at varying compression-ratios on petrol "A" and on benzene.

(2) Tests of volumetric efficiency over the whole available range of mixture strength with compression-ratios of 4 : 1 and 6 : 1.

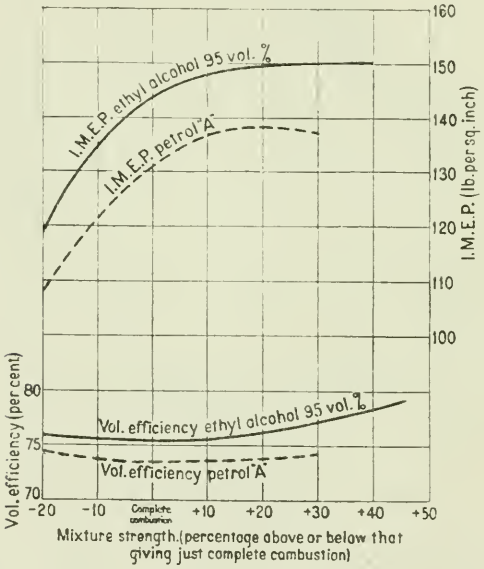


Fig. 34.—Curves showing variations of volumetric efficiency and I.M.E.P. with varying mixture strengths. (Compression-ratio constant at 5 : 1 and all other conditions constant.)

(3) Tests of volumetric efficiency with petrol "A" and with 95 Vol. per cent alcohol at a constant compression-ratio and constant heating.

(4) Tests of volumetric efficiency with constant mixture strength and varying heat input to the carburettor.

(5) Tests of volumetric efficiency with hot and cold cylinder walls at a constant compression-ratio and with other conditions constant.

When dealing with power output the author referred to the apparent variation in volumetric efficiency at different compression-

ratios, and pointed out that it was not at all easy to explain why this should occur. That it was the volumetric efficiency alone that varied was evident from the following considerations :—

(*a*) With increase of compression the thermal efficiency rose at a rate in excess of the rise in mean pressure.

(*b*) No mechanical or external temperature conditions were in any way affected by the change in compression-ratio.

The only possible inference to be drawn from these phenomena was that for some reason the volumetric efficiency became less as the compression-ratio was raised. The data obtained from recent tests with the air-measuring device referred to above confirm that this

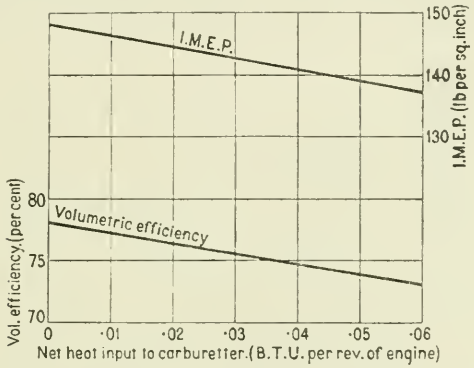


FIG. 35.—Curves showing variation of volumetric efficiency and I.M.E.P. with varying heat input to carburettor. (Compression-ratio constant at 60° C. Mixture strength 20 per cent richer than that giving just complete combustion. Fuel, petrol "A.")

is actually the case. From Fig. 36 it will be observed that direct air measurements under identical conditions show a steady fall in air consumption as the compression-ratio is raised. Every care was taken to ensure that neither the mixture strength, the temperature of the cylinder jackets, nor the rate of heat input to the carburettor was varied.

The curve shown in Fig. 33 illustrates the observed variation in volumetric efficiency over a wide range of mixture strength for petrol "A." As might be anticipated, the variation is small. It is seen that the volumetric efficiency is at a minimum when the mixture strength is that giving just complete combustion, and that it rises slightly as the mixture is either enriched or weakened from the normal : the former because of the increased proportion of liquid fuel entering the cylinder and the consequent fall in suction temperature due to the latent heat of evaporation, and the latter, presumably,

because of the lower surface temperature of the head, cylinder walls and piston resulting from the lower flame temperature at very weak mixtures. Fig. 34 shows the variation in indicated mean effective pressure and in volumetric efficiency with varying mixture strength for petrol "A" and for ethyl alcohol 95 Vol. per cent. It is seen that the increase in maximum power (as shown by the indicated mean effective pressure), comparing petrol and alcohol, is more than the increase of volumetric efficiency between these two fuels. This is probably to be explained by a fact, which has been established by previous tests, that the thermal efficiency with alcohol is slightly higher than with petrol. The rise in volumetric efficiency with rich mixtures is clearly shown in this Figure, particularly in the case of the curve for alcohol, indicating, as was suggested in the last para-

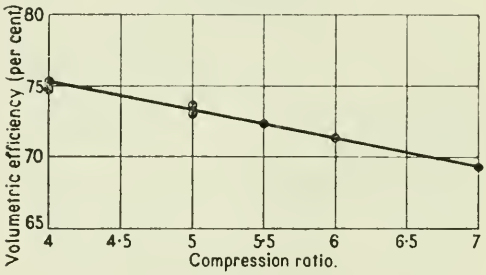


FIG. 36.—Curves showing variation in volumetric efficiency with different compression-ratios. (Constant mixture strength giving just complete combustion. Speed constant, 1500 revs. per minute. Jacket temperature constant, 60° C. Heating to air intake constant, 0.056 B.Th.Us. per rev. Fuel, petrol "A.")

graph, that the rise in volumetric efficiency and power is mainly due to the extra cooling of the charge by evaporation.

In Fig. 35 is shown the variation of volumetric efficiency and maximum indicated mean effective pressure when the heat input to the carburettor is varied. The relation between volumetric efficiency and heat input was found to be substantially the same for any of the fuels tested with the exception of the alcohols. The variations in temperature as recorded in the induction pipe due to differences in volatility of fuel appeared to have no influence on volumetric efficiency. The exception provided by the tests on ethyl alcohol confirms the conclusion arrived at previously by tests on power output, namely, that the whole of the liquid, with a fuel of very high latent heat, is not evaporated before the inlet valve closes, and the whole of the theoretical fall in suction temperature does not in practice occur.

In Fig. 37 is shown the change of volumetric efficiency with different temperatures in the water jacket of the cylinder. It will be seen that an increase in temperature of 60 degrees C. caused a fall of volumetric efficiency from 74.0 per cent to 72.2 per cent. Since the mean absolute suction temperature must have been approximately 370° C. it can be calculated from this result that a change of cylinder temperature of 60 degrees C. caused a change of suction temperature of approximately 12 degrees C.

Speaking generally, the tests at present carried out with the air-measuring device have revealed nothing new, but they have served to confirm the assumptions made previously as to the variations in volumetric efficiency with different fuels, different compression-ratios, and different temperature conditions. They have shown also that the apparatus may be relied on to give consistent results, and

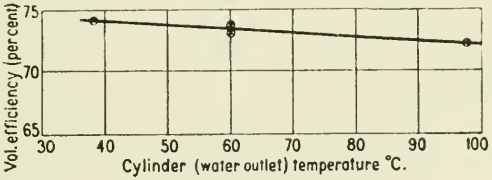


FIG. 37.—Curve showing variation of volumetric efficiency with change of cylinder temperature. (Compression 5 : 1. Other conditions constant as for Fig. 36.)

have demonstrated that certain phenomena which yet need investigation can be examined by the use of the variable-compression engine with this addition.

CONCLUSION.

The experimental results dealt with in the preceding pages give, as far as possible, a general summary of the experience gained during rather over two years' constant research.

It has not, of course, been possible to deal with more than a small proportion of the experiments carried out, but every effort has been made to select those which have led to the formation of definite conclusions, or which are at least suggestive. As in all such work, a very large proportion of the experiments was directed merely to confirming the results of previous tests; neither time nor space will permit of detailing the innumerable check tests carried out with one variable altered at a time, nor, indeed, would a relation of all such tests be of much general interest. It will, however, be well to point out that in all cases where a result was obtained which did

not at first sight appear to fall into line with the deductions formed at the time, check tests were taken progressively throughout the whole series of variables until the particular one, the influence of which was being exerted, had been traced. Except in a few instances where samples were very difficult to obtain, and only very small quantities were available, all the results quoted in the previous pages were checked again and again, and no figures have been published which have not been checked and repeated to within $\pm$ 0.5 per cent at least two or three times. In order that the reader may have the opportunity to judge for himself as to the accuracy of the readings obtained, and to draw his own conclusions as to the nature and possible influence of such disturbing factors as may remain, and which might affect the results, the apparatus used was described at the commencement of this volume.

TABLES.

TABLE I.—**Fuel Data for Some of**

Fuel.	A. Vapour tensions, 760 mm. Hg. and 0° C.	B. Viscosity at 20° C. C.G.S. units.	C. Latent heat of evaporation, B.Th.Us. per lb.	D. Calorific value, exclusive of latent heat (lower values). B.Th.Us. per lb.	E. B.Th.Us. per gal.	No
Petrols.						
Aromatic free	—	0.004	133	19,080	137,000	1
"A" petrol	28	0.005	142	18,450	144,300	2
"B" ,,	86	0.005	140	18,890	136,600	3
"C" ,,	54	0.005	135	19,000	138,100	4
"D" ,,	18	0.005	132	18,770	142,600	5
"E" ,,	70	0.005	133	18,970	136,400	6
"F" ,,	68	0.004	134	19,130	134,700	7
"G" ,,	44	0.005	—	—	—	8
,, H" ,,	17	0.006	145	18,790	144,100	9
Heavy Fuels.						
Heavy aromatic	—	0.007	136	17,900 (approx.)	158,500 (approx.)	10
Kerosene	—	0.019	108	19,000 (approx.)	154,400 (approx.)	11
Paraffin Series.						
Hexane (80%)	45	0.003	156	19,250	131,900	12
Heptane (97%)	11.5	0.004	133	19,300	132,800	13
Aromatic Series.						
Benzene (pure)	26	0.006	172	17,330 17,302	153,200 152,950	14
Toluene (99%)	9	0.006	151	17,580 17,522	153,000 152,500	15
Xylene (91%)	—	0.006	145	17,800	153,500	16
Naphthene Series.						
Cyclohexane (93%)	27.5	0.006	156	18,800	147,800	17
Hexahydrotoluene (80%)	—	—	138	18,760	146,200	18
Hexahydroxylene (60%)	—	—	133	18,770 (approx.)	139,700 (approx.)	19
Alcohol Group.						
Ethyl alcohol (98%)	12	0.012	406	11,450 11,480	91,400 91,600	20
Methyl alcohol (purified wood naphtha)	—	—	—	—	—	21
Acetone	—	0.003	239	12,350	98,440	22
Methylated spirits	—	0.010	(454)	(9,850)	(81,000)	23
Miscellaneous.						
Acetylene	Gas	—	—	21,010	—	24
CS_2 50% aromatic free 50% (by volume)	—	—	145	19,620	105,600	25
Ether (dilute)	185	0.002	(229)	(13,350)	(98,000) (approx.)	26

NOTE.—(1) Black figures are estimated from approximate composition.
(2) Where two values are given bracketed, the one underneath is the
(3) Final boiling point of petrol is always a somewhat uncertain

he Samples Tested (see pp. 59, 61 and 99).

No.	F. Sp. gr. at 15° C.	G (see p. 130). Boiling point (Engler distillation) per cent at 60° C.	80° C.	100° C.	120° C.	140° C.	160° C.	180° C.	H. Final deg. C.	J (see p. 130) Temp. rise °C. measured in induction pipe with std. heating.
1	0.718	1	16.0	49	72.0	85.0	93.0	—	—	+ 10.0
2	0.782	—	—	15	54.0	83.0	96.0	—	164	+ 11.0
3	0.723	4	37.5	79	99.0	—	—	—	126	+ 3.0
4	0.727	—	11.5	47	79.0	92.0	98.5	—	160	+ 7.0
5	0.760	—	—	13	66.0	89.0	97.5	—	166	+ 11.0
6	0.719	2	14.5	43	71.0	86.0	96.0	—	170	+ 10.5
7	0.704	1	27.0	65	86.5	94.5	—	—	153	+ 5.3
8	0.750	—	7.0	24	47.0	67.0	81.5	91	210	+ 18.0
9	0.767	—	—	7	55.0	83.0	94.0	—	176	+ 12.0
		160° C.	180° C.	200° C.	220° C.	240° C.	260° C.	280° C.		
10	0.885	8	30.0	50	65.0	77.0	90.0	—	275	+ 28.0
11	0.813	—	22.0	36	50.0	63.0	76.0	86	300	+ 31.0
		Range of boiling, ° C.								
12	0.685	40 to 88								0.0
13	0.691	98 to 98								+ 5.5
14	0.884	80 to 80								− 7.3
15	0.870	110 to 110 (approx.)								+ 8.0
16	0.862	84 to 143								+ 18.0
17	0.786	80.8 to 81								− 3.0
18	0.780	95.5 to 101.2								+ 3.0
19	0.744	103 to 123								+ 13.5
20	0.798	78								+ 1.5
21	0.829	—								− 8.5
22	0.798	56								—
23	0.821	—								− 1.0
24	—	—								
25	0.994	—								
26	0.735	35								

more recent, and is probably the more accurate determination.
quantity.

TABLE II.—Fuel

Fuel.*†	A. Approximate composition (p. 103). Paraffins, % by wt.	Aromatics, % by wt.	Naphths, % by wt.	D. In variable-compression en Compression at which deto occurred with maximum mixture and ignition timing Comp.-ratio.	Comp.-press., lb. per sq. in (gauge).	Comp.-press. temp., °C.	No.
Petrols.							
Ar. free ..	63.0	1.7	35.0	4.85	105.5	392	1
"A"	26.0	39.0	35.0	6.0	148.5	430	2
"B"	62.0	14.9	23.0	5.7	133.5	422	3
"C"	61.0	8.5	30.5	5.25	118.0	407	4
"D"	38.0	14.6	47.0	5.35	121.5	410	5
"E"	68.0	11.3	20.0	4.7	100.5	387	6
"F"	80.0	4.3	15.2	5.05	111.5	400	7
"G"	—	7.5	—	4.55	96.0	381	8
"H"	10.0	4.8	85.0	5.9	140.5	428	9
Heav. ar. ..	—	71.5	—	6.5	163.5	438	10
Ker.	—	—	—	4.2	86.0	369	11
Par.							
Hex.	77.0	2.7	20.0 app.	5.1	113.5	401	12
Hep.	100.0 to within	0.2%	—	3.75	72.0	353	13
Ar.							
Ben.	Negligible	98	Negligible	6.9*	179.0*	450*	14
Tol.	,,	99	,,	>7.0	>183.0	>452	15
Xy.	—	91	4	>7.0	>183.0	>452	16
Naph.							
Cy. hex. ..	—	4.6	93.0	5.9*	140.5*	427	17
Hex. tol. ..	—	10.0	78.0	5.8	136.5	425	18
Hex. xy. ..	—	—	60.0	4.9	107.0	394	19
Ol.	Olefines		Paraf. and naphth.				
Cracked ..	53.0	10.0	37.0	5.55	128.0	—	20
Al.	Water						
Eth. al. ..	1.5	—	—	>7.5	>204.0	>424	21
Meth. al. ..	Not fully determined			5.2*	116.5*	342*	22
Meth. sp. ..	7 approx.	—	—	6.5*	163.5*	382*	23
Acet.	0.5	—	—	>7.0	>183.0	—	24
Misc.			Arom. free by vol.				
Eth. 50% ..	2.5 app.	—	50.0	3.9	77.0	—	25
Eth.	5.0 app.	—	—	(2.95)	(47.5)	305	26
Carb. dis. 50%	—	—	50.0	5.15*	115.0	—	27
Carb. dis. ..	—	—	—	(5.4)	(123.0)	—	28

NOTES.—The sign * denotes that the value given for compression could not ( ) Where values are given in brackets, they are arrived at by these fuels with aromatic free petrol.

Column F.—The compression temperatures given in this column are to the latent heat of evaporation of the fuel on the one hand and a constant ND Signifies that there was no detonation up to the limit of pressure SS Cyclohexane did not behave normally in the supercharging engine. In the variable-compression engine exceeded those from detonation tests made All the naphthenes, however, were alike in that there was a greater tendency

† For full particulars of the fuels see Table I.

Characteristics (see pp. 59, 110).

	G.	H.	I.	J.	K.	L (p. 82, 103).	M.
	gine (p. 82) nation power (p. 87).	In supercharging engine (p. 82).		Ind. thermal effic'cy at highest useful compression, per cent.	Relative thermal efficiency (efficiency of 37.5% at compression ratio of 7 : 1 is taken as = 100), per cent.	Ind. mean pressure at highest useful compression with heat-input to carburettor of 65 B.Th. Us. per min., lb. per sq. in. (gauge).	Relative I.M.E.P. (the I.M.E.P. of 147 lb. per sq. in. obtained with toluene at 7 : 1 compression is taken as = 100), per cent.
No.	Tol. value, per cent.	Ind. mean pressure at which detonation occurred, lb. per sq. in. (gauge).	Tolu'ne value per cent (p. 84, 94).				
1	0.0	120.0	0.0	31.4	83.7	130.0	88.4
2	38.0	144.0	33.0	34.9	93.0	140.1	95.3
3	28.0	133.0	18.0	34.1	91.0	137.5	93.5
4	13.5	129.0	13.0	32.5	86.7	133.9	91.0
5	16.5	129.5	14.0	33.1	88.3	134.9	91.7
6	−5.0	117.0	−4.5	30.7	81.8	128.6	87.5
7	6.5	125.0	7.5	32.1	85.6	132.7	90.2
8	−10.0	115.0	−7.5	30.2	80.5	127.4	86.6
9	35.0	137.0	24.0	34.6	92.2	139.5	94.8
10	55.0	—	—	—	—	142.5	96.9
11	−22.0	—	—	—	—	123.0	83.6
12	8.0	124.5	7.0	32.4	86.4	133.1	90.5
13	−37.0	—	—	26.7	71.2	119.0	81.0
14	(67)	ND	(66)	37.2	99.2	146.5	99.6
15	100.0	ND	100.0	37.5	100.0	147.0	100.0
16	(85)	ND	(83)	37.3	99.5	146.8	99.9
17	35.0	148.0	38.088	34.9	93.0	139.2	94.6
18	31.5	140.0	28.0	34.3	91.5	137.9	93.7
19	1.5	121.5	2.5	31.5	84.0	130.0	88.4
20	23.5	- -	- -	33.9	90.4	136.0	92.5
21	>88.0	ND	(130)	40.4	107.9	156.5	106.4
22	—	—	—	35.0 ap.	93.3	146.6	99.6
23	—	—	—	38.5 ap.	102.7	155.5	105.8
24	>72.0	ND	(79)	—	—	—	—
25	−32.0	—	—	—	—	—	—
26	(—64)	—	—	(19.5)	52.0	(116.5)	79.2
27	—	—	—	—	—	—	—
28	(18)	(134.0)	(20)	—	—	—	—

be exceeded owing to pre-ignition occurring before audible detonation.
extrapolation of the curves obtained from tests made with mixtures of

calculated, taking into account the fall or rise in temperature due, respectively,
heat-input of 65 B.Th.Us. per minute on the other.
obtainable in the supercharging engine.
the case of the other samples of naphthenes the toluene values arrived at from
in the supercharging engine.
to pre-ignition than with the aromatics.

TABLE III.—**Fue**

Fuel.	A. Calorific (lower) value (exclusive of latent heat) (p. 96).		B. Calorific (lower) value (including latent heat at constant volume).		No.
	B.Th.Us. per lb.	B.Th.Us. per gallon.	B.Th.Us. per lb.	B.Th.Us. per gallon.	
Petrols.					
Ar. free	19,080	137,000	19,200	136,200	1
"A"	18,450	144,300	18,580	145,200	2
"B"	18,890	136,600	19,020	137,500	3
"C"	19,000	138,100	19,120	137,000	4
"D"	18,770	142,600	18,890	143,500	5
"E"	18,970	136,400	19,090	137,100	6
"F"	19,130	134,700	19,250	135,500	7
"G"	—	—	—	—	8
"H"	18,790	144,100	18,920	145,000	9
"I"	—	—	—	—	10
Heavy Fuels.					
Heavy ar.	17,900*	158,500*	18,030	159,600	11
Kerosene	19,000*	154,400*	19,100	155,200	12
Par.					
Pen. (normal) ..	19,600	122,300	19,740	123,100	13
Hex.	19,250	131,900	19,390	132,900	14
Hep.	19,300	132,800	19,420	134,100	15
Ar.					
Ben.	17,302	152,950	17,460	154,200	16
Tol.	17,522	152,500	17,660	153,600	17
Xy.	17,800	153,500	17,930	154,500	18
Naph.					
Cy. hex.	18,800	147,800	18,940	149,000	19
Hex. tol.	18,760	146,200	18,890	147,200	20
Hex. xy.	18,770*	139,700*	18,890	140,600	21
Ol.					
Cracked..	18,400*	139,400*	18,540	140,200	22
Al.					
Eth. al. (98.5%) ..	11,480	91,600	11,840	94,500	23
Eth. al. (95 Vol. %)	10,790	88.000	11,130	92.000	24
Meth. al.	9,630	79,900	10,030	83,300	25
Meth. sp.	10,200	83.700	10,580	86,900	26
But. al. (Coml.) ..	—	—	—	—	27
Ether (50% in Petrol)	16,700*	121,300*	16,830	122,500	28
Carb. dis. (50%) ..	10,600	105,400	10,730	106 600	29

Characteristics.

No.	C. Heat of combustion per cu. in. at n.t.p. of mixture giving complete combustion, ft.-lb.	D. Spec. volume after combustion ÷ that before combustion or "volume ratio."	E. Total energy liberated by combustion per cu. in. at n.t.p. of mixture giving complete combustion, ft.-lb. (pp. 96, 97, 99).	F. Air to fuel ratio by weight for complete combustion (p. 99).	G. Fall in temperature (calculated) of air-fuel mixture due to latent heat of evaporation, °C. (p. 99).
1	46.08	1.053	48.5	15.05	18.0
2	46.39	1.038	48.15	14.3	20.0
3	46.19	1.049	48.45	14.7	19.0
4	46.13	1.052	48.53	14.8	18.5
5	46.18	1.047	48.35	14.6	18.3
6	46.16	1.051	48.51	14.9	18.2
7	46.1	1.053	48.54	15.0	18.2
8	—	—	—	—	—
9	46.1	1.048	48.31	14.7	20.0
10	—	—	—	—	—
11	46.66	1.04	48.52	13.8	19.8
12	46.14	1.06	48.91	15.0	14.5
13	46.25	1.051	48.7	15.25	21.0
14	46.0	1.051	48.35	15.2	21.0
15	46.06	1.056	48.64	15.1	18.0
16	46.9	1.013	47.51	13.2	26.0
17	46.9	1.023	47.98	13.4	22.5
18	46.7	1.03	48.1	13.6	21.5
19	46.08	1.044	48.11	14.7	21.5
20	46.04	1.047	48.2	14.7	19.0
21	46.1	1.054	48.59	14.8	18.9
22	47.0	1.054	49.54	14.8*	20.8
23	44.5	1.065	47.36	8.9	85.0
24	44.0	1.065	46.86	8.4	98.0
25	45.5*	1.06*	48.2*	6.5	140.0
26	44.0	1.064	46.82	8.0*	110.0
27	—	—	—	—	—
28	46.4	1.06	49.2	13.0	22.0
29	40.2	0.98	39.4	10.8	27.0

*Approximately.

TABLE IV. (see p. 87),

Fuel.	A. Self-ignition temperature as determined by adiabatic compression with air (° C.).	B. Self-ignition temperature at atmospheric pressure as determined by Moore. With oxygen (° C.).	C. With air (° C.).
Aromatic free petrol	353	—	—
"A" petrol	367	—	—
"B" ,,	—	—	—
"C" ,,	—	—	—
"D" ,,	—	—	—
"E" ,,	—	—	—
"F" ,,	—	272	383
"G" ,,	—	—	—
"H" ,,	—	—	—
Heavy aromatics	—	—	—
Kerosene	—	252	—
Paraffin Series.			
Hexane, 80%	366	—	—
Heptane, pure	330	—	—
Aromatic Series.			
Benzene, 98%	419	566	—
Toluene, 99%	422	516 (90%)	—
Xylene, 91%	—	484 (Coml.)	—
Naphthene Series.			
Cyclohexane, 93%	387	—	—
Hexahydrotoluene, 78%	378	—	—
Hexahydroxylene, 60%	—	—	—
Alcohol Group and Miscellaneous.			
Ethyl alcohol, 98%	514	518	—
Methyl alcohol (purified wood naphtha)	457	—	—
Methylated spirits	—	—	—
Ether	256	190	347
Carbon disulphide	275	—	—

Column A.—The figures given in this column are the temperatures at which samples of the fuel were found just to ignite spontaneously when a mixture of fuel and air of a strength to give complete combustion was suddenly compressed, without initial turbulence, the ratio of compression being adjusted until self-ignition just occurred. This machine, which will be described later, has only recently been brought into operation and some uncertainty still exists as to the true value for the exponent of the compression curve. There is no reason, however, to doubt the relative accuracy of these figures.

Columns B and C.—The figures given in these columns are taken from Mr. Harold Moore's paper on "The Spontaneous Ignition Temperatures of Liquid Fuels for Internal Combustion Engines," read before the Society of Chemical Industry, December 1st, 1916.

TABLE V. (see pp. 100, 103).

Name of fuel.	A. Total energy liberated by combustion, ft.-lb. per std. cu. in. (Col. E of Table III.)	B. Relative power output allowing for increase in density due to evaporation. Octane = 100.
Paraffin Series.		
Hexane	48.33*	100.2
Heptane	48.64	100.1
Octane	48.73	100.0
Nonane	48.78	—
Decane	48.82	99.4
Aromatic Series.		
Benzene	47.51	100.1
Toluene	47.98	100.0
Xylene	48.26	100.6
Naphthene Series.		
Cyclohexane	48.11	100.0
Hexahydrotoluene	48.32	99.8
Hexahydroxylene	48.49	99.8
Olefine Series.		
Heptylene	49.1	102.5
Decylene	—	—
Alcohol Group.		
Ethyl alcohol	47.4	120.0
Methyl alcohol	45.05	143.0
Miscellaneous.		
Ether	49.98	107.0
Carbon disulphide	34.5	74.0
Acetylene	51.85	101.0
Carbon monoxide	39.28	77.0
Hydrogen	33.9	66.0

* In certain cases variations will be found between the figures given in different Tables under the same heading, due to the fact that the results are taken from from different tests. The differences are, however, so small as to be negligible.

TABLE VI. (see p. 103).

Fuel.	A. Calculated figure of merit for power output. Octane = 100.	B. With no heat to carburettor. Room temperature 15° C. Compression-ratio 5 : 1. Ind. mean pres., lb. per sq. in.	C. With no heat to carburettor. Room temperature 15° C. Compression-ratio 5 : 1. Temp. ° C. recorded in inlet-v. housing.	D. With 65 B.Th.Us. per minute added. Room temperature 15° C. Comp.-ratio 5 : 1. Ind. mean pres., lb. per sq. in.	E. With 65 B.Th.Us. per minute added. Room temperature 15° C. Comp.-ratio 5 : 1. Temp. ° C. inlet-v housing.	F. With constant recorded temp. of 15° C. inlet housing. Room temperature 15° C. Comp. 5 : 1. Ind. mean pres., lb. per sq. in.	G. With constant recorded temp. of 15° C. inlet housing. Room temperature 15° C. Comp. 5 : 1. Added heat required, B.Th.Us. per min.
Petrols.							
Ar. free	100.0	139.5*	10.0	131.3*	25.0	136.5*	24.7
"A"	99.9	139.6	11.0	131.2	26.0	137.2	17.3
"B"	99.7	139.8	7.0	131.5	18.0	133.9	46.0
"C"	99.7	139.5	8.5	131.0	22.0	135.0	31.2
"D"	99.9	139.7	10.5	131.2	26.0	137.1	17.3
"E"	99.7	139.5*	11.0	131.0*	25.5	136.3*	22.5
"F"	100.3	139.9	8.0	131.8	20.3	135.2	37.0
"G"	—	139.6*	15.0	131.5*	33.0	139.5*	—
"H"	99.7	139.4	11.5	131.0	27.0	137.2	14.3
Heavy ar.	100.1	—	—	131.1	43.0	—	—
Kerosene	99.5	—	—	130.7*	46.0	—	—
Par. Series.							
Hexane	100.3	140.1	−2.5	132.3	15.0	132.3	65.0
Heptane	99.9	139.5*	8.5	131.2*	20.5	135.5*	34.7
Ar. Series.							
Benzene, 98%	100.1	139.8	—	131.6	7.7	123.7	130.0
Toluene, 99%	100.0	139.5	11.0	131.5	23.0	137.1	21.7
Xylene, 91%	100.3	139.3	17.0	131.5	33.0	—	—
Napth Series.							
Cy. hex., 93%	100.0	139.6	−1.0	131.3	12.0	129.5	80.0
Hex. tol., 78%	99.7	139.5	5.5	131.0	18.0	132.8	49.0
Hex. xy., 60%	99.7	139.0*	13.0	130.8*	28.5	138.0*	7.8
Olefine Series.							
Cracked spirit	101.5	139.8	13.0	131.6	25.5	138.0	13.0
Alcohol Group.							
Ethyl alcohol	120.8	135.2	9.5	137.8	16.5	139.3	52.0
Methyl alcohol†	—	151.7	−1.0	144.8	6.5	135.5	152.0
Meth. spirits	132.6	151.3	6.5	144.5	14.0	143.2	76.0
Miscellaneous.							
Ether, 50%	103.2	144.3*	4.0	136.0*	16.0	136.5*	60.0
Carb. dis., 50%	83.2	135.0	−4.0	124.5	8.0	—	—

NOTE.—* Indicates that owing to detonation occurring at a compression-ratio 5 : 1 the value given is calculated from the observation made when running at the highest usable compression.

† Purified naphtha.

TABLE VII. (see p. 113).

Fuel.	A. Calorific value "lower" (exclusive of latent heat), B.Th.Us. per lb.	B. Minimum consumption at 5 : 1 comp., lb. per I.H.P. per hour.	C. Thermal efficiency from Cols. (A) and (B), per cent.	D. Latent heat at const. pressure (atmospheric), B.Th.Us. per lb.	E. Corrected "lower" calorific value, B.Th.Us. per lb. (p. 121).	F. Corrected thermal efficiency from Cols. (B) and (E), per cent.
Petrols.						
Ar. free ..	19,080	0.415*	32.1	133	19,200	31.9
"A"	18,450	0.432	31.9	142	18,580	31.7
"B"	18,890	0.423	31.9	140	19,020	31.7
"C"	19,000	0.421	31.8	135	19,120	31.6
"D"	18,770	0.422	32.1	132	18,890	31.9
"E"	18,970	0.421*	31.9	133	19,090	31.7
"F"	19,130	0.414	32.1	134	19,250	31.9
"H"	18,790	0.425	31.9	145	18,920	31.7
Heavy ar. ..	17,900	0.510	27.8†	136	18,030	27.6†
Kerosene ..	19,000	0.523*	25.6†	108	19,100	25.4†
Paraffin.						
Hexane ..	19,250	0.411	32.2	156	19,390	32.0
Heptane ..	19,300	0.410*	32.1	133	19,420	31.9
Aromatic.						
Benzene ..	17.300	0.458	32.1	172	17,460	31.8
Toluene ..	17.520	0.455	31.9	151	17,660	31.7
Xylene	17,800	0.452	31.6	145	17,930	31.4
Naphthene.						
Cyclohexane .	18,800	0.420	32.1	156	18,940	31.9
Hex. tol. ..	18,760	0.425	31.9	138	18,890	31.7
Hex. xy. ..	18,770	0.424*	32.0	133	18,890	31.8
Olefine.						
Cracked.. ..	18,400§	0.429	32.3	150§	18,540	32.0
Alcohol.						
Eth. al.,98.5%	11,470	0.663	33.5	406	11,840	32.4
,, ,, 95 Vol.%	10,790	0.705	33.5	442	11,130	32.5
Meth. al. ..	9,630	0.777	34.0	500§	10,030	32.7
Meth. sp. ..	10,200‡	0.740	33.7	450§	10,580	32.5

NOTES.—* This sign indicates that at a compression-ratio of 5 : 1 the fuel could not be used owing to detonation. These values for fuel consumption are therefore calculated from the readings obtained from tests made with lower compressions. The close agreement between thermal efficiencies calculated from these results clearly shows the uniformity of the relation between compression-ratio and thermal efficiency for the different fuels tested.

† The low thermal efficiency observed for these fuels is quite certainly due to non-combustion of a portion of the fuel, which probably remains liquid during the working stroke.

‡ The calculated heat value for methylated spirits was given earlier as 9850 under a misapprehension as to the water content. The value now given, namely, 10,200 B.Th.Us. per lb., is probably nearly correct for the ordinary purple variety of this specific gravity.

§ Approximately.

TABLE XII.—**Experimental Results**

Fuel.	A (p. 124). Minimum consumption at compression-ratio of 5 : 1 per I.H.P. per hour.		B (p. 124). Minimum consumption at highest useful compression per I.H.P. per hour.		No.
	lb.	Pints.	lb.	Pints.	
Petrols.					
Aromatic free petrol	0.415*	0.462*	0.422	0.471	1
"A" petrol	0.432	0.442	0.393	0.402	2
"B" ,,	0.423	0.468	0.393	0.435	3
"C" ,,	0.421	0.463	0.410	0.451	4
"D" ,,	0.422	0.445	0.407	0.428	5
"E" ,,	0.421*	0.469*	0.435	0.484	6
"F" ,,	0.414	0.471	0.412	0.469	7
"G" ,,	0.426	0.454	0.449	0.478	8
"H" ,,	0.425	0.443	0.389	0.405	9
"I" ,,	0.418*	0.460*	0.457	0.503	10
Heavy Fuels.					
Heavy aromatics	0.510	0.461	0.447	0.404	11
Kerosene	0.523*	0.515*	0.581	0.571	12
Paraffin Series.					
Pentane (normal)	—	—	—	—	13
Hexane (80% pure)	0.411	0.480	0.405	0.473	14
Heptane (97% pure)	0.410*	0.475*	0.491	0.568	15
Aromatic Series.					
Benzene (pure)	0.458	0.415	0.392	0.355	16
Toluene (99% pure)	0.455	0.418	0.385	0.354	17
Xylene (91% pure)	0.452	0.420	0.381	0.354	18
Naphthene Series.					
Cyclohexane (93% pure)	0.420	0.427	0.385	0.392	19
Hexahydrotoluene (80%)	0.425	0.436	0.394	0.404	20
Hexahydroxylene (60%)	0.424*	0.456*	0.429	0.461	21
Olefines.					
Cracked spirit (53% unsat.)	0.429	0.453	0.405	0.428	22
Alcohol Group, etc.					
Ethyl alcohol (98%)	0.663	0.665	0.532	0.533	23
Ethyl alcohol (95 Vol. %)	0.705	0.692	0.565	0.555	24
Methyl alcohol (wood naphtha)	0.777	0.750	0.725	0.700	25
Methylated spirits	0.740	0.721	0.625	0.609	26
Butyl alcohol (coml.)	0.566	0.550	0.472	0.459	27
Ether (50% in petrol)	—	—	—	—	28
Carbon disulp. (50%)	—	—	—	—	29

NOTES.—* This sign indicates that the values are only calculated since these
The values have been inserted to show the efficiency and power
‡ Approximately.

for Variable-compression Engine.

No.	C. Thermal efficiency at compression-ratio of 5 : 1, per cent.	D. Max. ind. mean effective pressure at compression-ratio of 5 : 1. Heat 65 B.Th.Us. per min., lb. per sq. in.	E (p. 130). Rise or fall of temp. in induction pipe (indicating app. mean volatility). Heat 65 B.Th.Us. per min., ° C.	F. Relative thermal efficiency (compared with that obtained with toluene) at the same compression-ratio, per cent.	G. Relative I.M.E.P. (compared with that obtained with toluene) at the same compression-ratio, per cent.
1	31.9*	131.3*	+ 10.0	100.0‡	100.0‡
2	31.7	131.2	+ 11.0	100.0‡	100.0‡
3	31.7	131.5	+ 3.0	100.0‡	100.0‡
4	31.6	131.0	+ 7.0	100.0‡	100.0‡
5	31.9	131.2	+ 11.0	100.0‡	100.0‡
6	31.7*	131.0*	+ 10.5	100.0‡	100.0‡
7	31.9	131.8	+ 5.3	100.0‡	100.0‡
8	—	131.5*	+ 18.0	—	100.0‡
9	31.7	131.0	+ 12.0	100.0‡	100.0‡
10	—	131.7*	+ 14.0	—	100.0‡
11	27.6	131.1	+ 28.0	86.0	100.0‡
12	25.4*	130.7*	+ 31.0	80.0	99–100
13	—	131.3	—	—	100.0‡
14	32.0	132.3	0	100.0‡	100.0‡
15	31.9*	131.2*	+ 5.5	100.0‡	100.0‡
16	31.8	131.6	− 7.3	100.0‡	100.0‡
17	31.7	131.5	+ 8.0	100.0‡	100.0‡
18	31.4	131.5	+ 18.0	100.0‡	100.0‡
19	31.9	131.3	− 3.0	100.0‡	100.0‡
20	31.7	131.0	+ 3.0	100.0‡	100.0‡
21	31.8*	130.8*	+ 13.5	100.0‡	100.0‡
22	32.0	131.6	+ 10.5	100.0‡	100.0‡
23	32.4	137.8	+ 1.5	102.0	105.0
24	32.5	142.0	+ 2.0	102.5	108.0
25	32.7	144.8	− 8.5	103.0	110.0
26	32.5	144.5	− 1.0	102.5	110.0
27	—	138.0	+ 10.0	—	105.0
28	—	136.0*	+ 1.0	—	103.5
29	—	124.5	− 7.0	—	94.7

fuels could not be tested at a compression-ratio of 5 : 1 owing to detonation.
obtained relatively to the other fuels if used at the same compression.

TABLE VIII. (see p. 122).

A.	B.	C.	D.	E.
Compression-ratio.	Ind. therm. efficiency, alcohol, 95 vol. per cent.	Ind. therm. efficiency, hydro-carbons.	Relative efficiency, alcohol, 95 vol. per cent.	Relative efficiency, hydro-carbons.
3.8 : 1	26.8	26.3	65.0	63.7
5.0 : 1	32.5	31.7	68.5	66.8
7.0 : 1	38.2	37.4	70.7	69.3

TABLE IX. (see p. 123).

Fuel.	Total heat of fuel, B.Th.Us. per hour.	Heat to I.H.P. B.Th.Us. per hour.	Heat to jacket water, B.Th.Us. per hour.	Heat to exhaust, etc., B.Th.Us. per hour.
Petrol	253,000	66,800	72,400	113,800
Alcohol, 95 vol. per cent.	275,000	73,900	67,200	133,900

TABLE X. (see p. 126).

	A.	B.					C.	D.
		Engler distillation per cent at						
Fuel.	S.G. at 15° C.	60° C.	80° C.	100° C.	120° C.	140° C.	Inlet temp. with std. heating, ° C.	Vapour pressure at zero C. (mm. of mercury).
Petrol "F" ..	0.704	1	27	65	86.5	94.5	20.3	68.0
Heptane ..	0.691	Boils at 98° C.					20.5	11.5

TABLE XI. (see p. 126).

	A.	B.	C.	D.
Fuel.	Boiling point ° C.	Vapour pressure at zero C. mm. mercury.	Latent heat B.Th.Us. per lb.	Air/fuel ratio by weight.
Hexane	69	45.0	156	15.20
Cyclohexane	81	27.5	155	14.70
Benzene	80	26.0	172	13.20
Ethyl alcohol	78	12.7	397	8.95
Ether	35	185.0	158	11.14
Heptane (pure)	98	11.5	133	15.10

SECTION III.

REPORT OF THE EMPIRE MOTOR FUELS COMMITTEE.

TESTS ON 95 VOL. PER CENT ALCOHOL.*

The following description relates to the series of tests on 95 Vol. per cent alcohol undertaken for the Empire Motor Fuels Committee. An outline of the scope of these tests was given at the commencement of this volume (see p. 62 *ante*). The research apparatus used was substantially the same as that employed for the tests already described, with the exception of the new head fitted to the variable-compression engine (see p. 135).

Note on the Calorific Value of Alcohol Sample.

When computing the thermal efficiency obtained with alcohol we have taken the lower effective calorific value of the fuel used as 11,130 B.Th.Us. per lb., or 11.330 B.Th.Us. per pint at 15° C., and we have arrived at this calorific value as follows :—

The most reliable and up-to-date estimates of the calorific value of pure ethyl alcohol as obtained in a bomb calorimeter give the lower value, after deducting the latent heat of evaporation of the water formed, as 11,640 B.Th.Us. per lb. The sample used for these tests contained 7.3 per cent by weight of water, so that the lower heat value of the liquid becomes 10,790 B.Th.Us. per lb. When the heat value is determined by bomb calorimeter a proportion of the heat of combustion of the vapour is absorbed in overcoming the latent heat of evaporation of the liquid fuel. When used in an engine, however, the whole of the heat of evaporation is supplied in part by preheating the fuel before its entry to the cylinder, and in part by contact with the hot cylinder walls and still hotter residual exhaust products, while any liquid still remaining is evaporated during the compression stroke ; hence it is safe to assume that the whole of the heat required for evaporation has been supplied before combustion takes place, that is to say, outside the thermal cycle of the engine. In correcting the bomb calorimeter results, the latent heat of evaporation at constant volume must be taken into account, and this is approximately 94 per cent of the latent heat at constant pressure as usually determined. We have therefore taken the effective lower calorific value as equal to the bomb calorimeter reading plus 94 per cent of the latent heat of evaporation of the fuel (*i.e.*, plus 340 B.Th.Us. per lb.), or 11,130 B.Th.Us. per lb net. In arriving at the effective calorific value of the petrol used for check

* Throughout the test results which follow, the strength of alcohol is always expressed in Volumes per cent.

tests we have followed the same reasoning, *i.e.*, we have added to the measured figure of the heat of combustion 94 per cent of the latent heat of evaporation of the fuel, and on this determination the effective calorific value of "A" petrol works out at 18,200 B.Th.Us. per *pint* at 15° C.

In all cases the fuel has been measured by volume and corrected for a temperature of 15° C.

Explanation of Test Sheets.

The readings given on the test sheets, pp. 163 to 191, are those actually recorded by the observers during the test, or are calculated from them. The columns, reading from left to right, give the following data :—

Column 1.—The revs. per minute quoted in the first column represent the actual number of revolutions recorded by the magnetically-operated counter during the consumption of 0.25 pint of fuel and corrected to 60 seconds. The speed of the engine is maintained approximately constant by means of the hand-operated rheostat on the dynamometer field circuit, but since the precise number of revolutions is automatically recorded, variations in speed do not affect the accuracy of observation, for, as will be seen from the test results, the efficiency in all cases is virtually constant over a very wide range of speed.

Column 2.—The figures given in this column denote the readings of the spring balance on the dynamometer torque arm.

Column 3.—The figures given in this column represent the actual torque exerted at a radius of 36in., and are arrived at by deducting the reading of the spring balance from that of the dead weight on the torque arm, the latter being given above the second and third columns.

Column 4.—The figures given in this column represent the mean effective pressure exerted on the piston, and are calculated from the measured torque and the internal friction, which latter has been ascertained frequently under varying conditions of temperature, etc., and which, as explained previously, is checked during or at the conclusion of every test. In all cases the indicated mean pressure has been corrected for a standard barometer reading of 29.9in. Hg. The indicated mean pressure affords the most accurate and convenient expression of the power developed, since it is not influenced by variations in speed.

Column 5.—The indicated horse-power quoted in this column is the mean indicated horse-power developed during the consumption of the measured quantity of fuel, and is arrived at

from the torque, revolutions and time ; these figures, therefore, are not corrected for barometer. The indicated horse-power figures quoted in the heat-balance sheets are the same as on the test sheets, but corrected for speed variations.

Column 6.—The water outlet temperature is measured by means of a Centigrade thermometer fitted in the water outlet passage from the cylinder head.

Column 7.—The inlet housing temperature is taken from the readings of a Centigrade thermometer fitted in an insulated pocket projecting well into the induction pipe close to its junction with the cylinder head. As explained previously, these readings are of doubtful value so far as their absolute accuracy is concerned, but relatively they are very instructive. For example, in the directly comparable tests on petrol and alcohol in Series I. at 1500 revs. per minute, when the heat-input to the carburettor was precisely the same in both cases, the temperature readings were 26.0° C. and 16.0° C., the temperature of the surrounding air, in both cases, being approximately 14° C., showing a rise in temperature with the same heat-input of 12 degrees C. and 2 degrees C. respectively.

Column 8.—These temperatures are taken from a thermometer suspended near the air intake to the carburettor.

Column 9.—The ignition settings refer to the readings of the long pointer moving over a dial attached to the contact-breaker, and the precise point at which contact is broken, when the pointer is at 3 on the dial, is stated at the head of each test sheet. Each division on the dial represents 8 degrees of the crankshaft ; thus if contact is broken at 28 degrees before top dead-centre when the pointer is at 3, this will occur 36 degrees early when the pointer is at 4.

Column 10.—This represents the number of seconds that elapse during the consumption of the measured quantity of fuel.

Column 11.—This gives the fuel consumption in terms of pints per indicated horse-power per hour calculated from Columns 5 and 10.

Column 12.—In this column is recorded the voltage across the heater coils in the air-intake passage to the carburettor. Since the resistance of these coils is constant, the heat-input is proportional to the square of the voltage. In all cases the voltage is adjusted to give a heat-input proportional to the speed of the engine and amounting to 0.0433 B.Th.Us. per revolution per minute (except in Series IV., in which the heat-input was increased to 0.066 B.Th.Us. per revolution per minute), a rate of heat-input which corresponds very approximately to that provided by an average water-warmed induction system.

The brake horse-power (B.H.P.) and brake mean pressure (Pb) developed during any test are given by the following formulæ :—

$$\text{B.H.P.} = w\,n \times 0.000571,$$
$$\text{Pb} = 3.556\,w,$$

where w equals the net weight on the torque arm as given in Column 3 of the tests sheets.

Owing to the very large valve area provided, the mean pressure developed in this engine rises with increase of speed and only reaches a maximum at about 2000 revs. per minute, corresponding to a piston speed of 2666ft. per minute. The rise in torque with increase of speed is not truly progressive over the whole range, but follows a slightly waved line, as shown in Fig. 43, p. 197, due to resonance in the exhaust pipe. For the same reason it is difficult to carry out steady tests at any speed below about 975 revs. per minute on account of the steeply-rising torque curve, which renders the speed control rather troublesome. Owing to the resonance effects in the exhaust pipe, both the power output and efficiency of the engine are very slightly below the true average at 1500 revs. per minute. This, however, has the effect of flattening out the torque curve round about this speed, and so both renders the speed control easier, and at the same time further diminishes any slight variations in torque or efficiency due to changes of speed.

Notes on the Test Sheets and Curves.

In all the test sheets and curves, unless otherwise stated, the performance of the engine both as regards power output and efficiency is expressed in terms of the *indicated* horse-power, mean pressure or consumption. Experience has shown that when expressed in these terms the results are more readily applicable to other engines of which the mechanical efficiencies vary so widely as frequently to mask small differences, and so render the result misleading.

When applying the data obtained to other engines the following circumstances must be borne in mind :—

(1) That the engine used for these experiments has a quite exceptionally high thermal efficiency.

(2) That its mechanical efficiency also is very high indeed.

(3) That owing to the very low piston friction, it is far less susceptible to changes in the temperature of the circulating water than ordinary engines.

(4) That it has one cylinder only, and that therefore the problems of distribution play very little part in its performance.

On some of the test sheets the check results obtained on petrol or benzole are recorded, together with the observed consumption. It should be emphasised that the figures found for power and con-

sumption during the check tests do not necessarily represent either the maximum power obtainable or the maximum economy; they represent merely the power obtained at a certain consumption of fuel and are compared with standard curves for that fuel. In most cases the check tests were run on alcohol also, and the standard fuels were used only when the conditions were such that no previous data were available with alcohol. In these check tests the power and consumption under certain speed and temperature conditions, etc., must be within ± 0.5 per cent of that obtained during the calibration tests of the engine on the same standard fuel and under the same conditions. Check tests taken at the commencement of a series of tests are expected to show about 0.5 per cent less power and greater consumption than the standard calibration-tests, while those taken at the end of a series when the engine is thoroughly warmed up should be equal to, or may be very slightly superior to, the standard figures.

At any given speed the power output and consumption are expressed in terms of indicated mean pressure plotted on a horizontal scale, and fuel consumption on a vertical scale. The curves so obtained are generally similar in form in all cases and on all fuels, but it will be observed that on alcohol the power increases slightly as the mixture is enriched, and that this increase extends over a very much greater range on alcohol than on petrol or benzole.

Deviations from Test Schedule.

In some instances it has been found necessary to deviate very slightly from the schedule of tests agreed upon with Dr. Ormandy and Mr. Watson, but such deviations as we have made do not affect the programme materially.

Series I. called for a series of power and consumption tests over the whole range of mixture strength at compression-ratios of 3.8 : 1 and 7 : 1 and at piston speeds of 800, 1200, 1600 and 2000ft. per minute. We found that we could not obtain really accurate readings at a piston speed of 800ft. per minute, partly because of the natural characteristics of the engine, which has a steeply-rising torque curve at about this speed, and partly because we could not supply sufficient field excitation to the dynamometer at so low a speed and still retain a sufficient reserve for adjustment and control, particularly when running with the high compression. After some preliminary runs we decided that it would be preferable to run the tests at mean speeds of 975, 1300, 1500 and 1675 revs. per minute, corresponding to piston speeds of 1300, 1733, 2000 and 2233ft. per minute respectively. These speeds were selected because they were found to be the most convenient from the point of view of dynamometer control.

Series II.—For similar reasons we ran this series at piston speeds of 1250 and 2000ft. per minute, as against 1200 and 2000ft. per minute

in the schedule. We did not carry out a special check test on petrol at the lower speed, because we found that we had ample data relating to the "A" petrol under these conditions, from which we were able to verify that the behaviour of petrol and alcohol was precisely the same at the lower speed as at the higher speed.

Series II. tests were all run with a compression-ratio of 5.45 : 1 instead of 5 : 1, as originally agreed. Our objects in using the higher compression-ratio were :—

(*a*) This is the highest compression at which any "*petrol*" now on the market can be used in this engine.

(*b*) We have run other and similar tests of several fuels at this ratio, and therefore have a large amount of comparative data available.

Series III.—We had not sufficient water supply to reduce the jacket temperature to 30° C. under all conditions, and therefore had to run at 35° C. for the cold-jacket tests. Also, we could not maintain 90° C. in the jackets without local boiling, and an irregular flow; we therefore reduced the temperature to 85° C., at which a perfectly steady flow could be maintained.

Series IV.—We considered it more informative to express the heat input to the carburettor in terms of B.Th.Us. than of watts. We therefore ran this series with a heat-input of 100 B.Th.Us. per minute instead of 2000 watts; the two rates are very nearly the same, but we thought it desirable, while changing the unit, to adhere to round numbers where possible.

In all other respects we have adhered strictly to the schedule of tests agreed upon and set forth on p. 62 *ante*, and the complete results are given in the following pages.

SERIES I. TEST No. 1.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 15°C. *Barometer* 30.22in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1484	7.0	33.0	134.5	32.0	58.8	14.0	13.0	3.6	25.2	1.115	87.5
1484	6.9	33.1	135.0	32.3	59.7	14.0	13.0	3.6	27.0	1.03	,,
1500	7.4	32.6	133.5	32.4	61.6	14.2	13.3	3.6	29.6	0.937	,,
1492	8.0	32.0	131.0	31.5	63.1	14.8	13.3	3.6	32.0	0.88	,,
1484	8.5	31.5	129.5	31.0	64.8	15.0	13.4	4.0	33.8	0.858	,,
1492	9.2	30.8	127.0	30.6	65.3	15.2	13.5	4.0	35.1	0.838	,,
1500	10.2	29.8	124.0	29.8	65.0	15.3	13.6	4.3	36.0	0.838	,,
1496	11.0	29.0	121.0	29.0	64.3	15.5	13.7	4.5	37.1	0.835	,,
1500	12.0	28.0	117.5	28.3	63.5	15.5	13.8	4.5	37.8	0.842	,,
1492	14.7	25.3	108.0	25.9	61.8	15.5	14.0	5.0	40.5	0.857	,,
1505	4.9	4.9	17.4	—	61.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 3.8 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
32.3	404,000	81,500	60,200	20.2	14.9	64.9
32.6	384,000	82,300	60,800	21.5	15.8	62.7
32.4	344,000	82,500	63,800	23.9	18.6	57.5
31.6	314,000	80,300	65,300	25.6	20.8	53.6
31.4	300,000	79,000	67,600	26.3	22.5	51.2
30.8	291,000	78,000	67,900	26.8	23.3	49.9
29.8	283,000	76,000	68,400	26.9	24.2	48.9
29.2	275,000	73,900	67,200	26.9	24.4	48.7
28.3	270,000	72,200	66,000	26.7	24.5	48.8
26.0	252,500	66,100	63,400	26.2	25.1	48.7

SERIES I. TEST NO. 2.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 15°C. *Barometer* 30.27in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb. Spring balance	Nett weight	I.M. E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
1288	7.8	32.2	130.3	27.1	51.3	15.7	14.2	3.5	26.6	1.25	82
1288	7.9	32.1	130.1	27.05	57.8	15.5	14.2	3.5	26.6	1.25	,,
1304	8.0	32.0	129.8	27.2	59.4	15.5	14.2	3.5	27.5	1.20	,,
1284	8.0	32.0	129.8	26.8	60.5	15.7	14.1	3.5	29.0	1.16	,,
1292	8.3	31.7	128.2	26.7	62.4	16.0	14.1	3.5	33.0	1.02	,,
1308	8.8	31.2	127.5	26.7	65.2	16.2	14.1	3.7	36.3	0.93	,,
1284	9.8	30.2	124.0	25.4	67.6	16.6	14.0	3.8	41.0	0.865	,,
1288	11.1	28.9	119.5	24.6	68.5	17.0	14.2	4.2	44.5	0.821	,,
1286	12.6	27.4	113.5	23.4	68.4	17.2	14.5	4.5	46.7	0.825	,,
1290	15.0	25.0	105.0	21.7	66.7	17.2	14.8	4.6	49.3	0.840	,,
1300	4.45	4.45	—	—	65.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 3.8 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1300 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
27.3	385,000	69,200	—	18.0	—	—
27.25	384,000	69,000	47,800	18.0	12.5	69.5
27.15	371,000	69,400	49,600	18.6	13.4	68.0
27.05	352,500	68,400	50,700	19.4	14.4	66.2
26.9	306,000	68,100	52,100	22.3	17.0	60.7
26.6	293,000	68,100	55,000	23.3	18.8	57.9
25.7	252,500	64,800	56,800	25.7	22.6	51.7
24.9	229,000	62,800	58,000	27.4	25.3	47.3
23.6	219,000	59,700	57,000	27.3	26.0	46.7
21.9	208,000	55,400	56,000	26.9	26.9	46.2

SERIES 1. TEST NO. 3.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 15·4°C. *Barometer* 30.29in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature °C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ½ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
1664	6.5	33.5	138.5	37.0	57.2	15.5	14.9	3.8	24.2	1.00	92
1660	6.7	33.3	137.5	36.7	57.8	15.5	15.0	4.0	26.6	0.922	,,
1660	7.6	32.4	134.5	35.9	59.5	15.5	15.0	4.0	29.5	0.850	,,
1672	9.0	31.0	129.5	34.8	59.8	15.7	15.0	4.2	31.0	0.835	,,
1674	10.9	29.1	122.5	33.0	58.3	15.5	15.0	4.6	33.0	0.825	,,
1670	13.0	27.0	115.0	31.1	56.3	15.3	15.0	5.0	34.8	0.830	,,
1680	14.6	25.4	109.5	29.4	54.8	15.5	15.0	5.3	35.4	0.865	,,
1675	5.3	5.3	—	—	55.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 3.8 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1675 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
37.3	418,000	94,200	72,500	22.5	17.3	60.2
37.0	382,000	93,400	73,900	24.4	19.3	56.3
36.2	345,000	91,400	76,400	26.5	22.1	51.4
34.9	329,500	88,600	78,000	26.9	23.7	49.4
33.0	308,000	84,600	75,500	27.3	24.5	48.2
31.2	292,000	79,200	75,100	27.1	25.1	47.8
29.3	287,000	74,900	69,500	26.1	24.2	49.7

SERIES I. TEST No. 4.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 15·6°C. *Barometer* 30.29in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature °C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
996	9.4	30.6	122.5	19.6	58.7	17.0	15.0	3.0	36.2	1.27	72
984	9.5	30.5	122.2	19.3	58.9	17.5	15.0	3.0	37.1	1.25	,,
985	9.6	30.4	121.8	19.2	59.1	17.7	15.0	3.0	39.2	1.195	,,
988	9.7	30.3	121.5	19.2	61.0	17.9	15.2	3.0	41.8	1.112	,,
986	9.9	30.1	121.0	19.1	61.8	18.2	15.3	3.2	44.0	1.070	,,
980	10.4	29.6	118.5	18.7	64.6	18.5	15.3	3.5	52.3	0.920	,,
980	11.5	28.5	115.0	18.1	67.2	18.5	15.1	3.5	59.2	0.840	,,
975	13.9	26.1	106.5	16.6	67.7	18.3	15.0	4.0	65.0	0.835	,,
982	15.0	25.0	102.5	16.1	66.3	18.2	14.8	4.0	66.4	0.840	,,
975	3.85	3.85			66.0			—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 3.8 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 975 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
19.2	281,000	50,000	38,000	17.8	13.5	68.7
19.1	273,000	49,200	38,300	18.0	14.0	68.0
19.0	259,000	49,000	39,500	18.9	15.2	65.9
18.9	243,000	49,000	41,000	20.2	16.8	63.0
18.8	231,000	48,700	41,600	21.1	18.0	60.9
18.6	194,500	47,800	43,600	24.6	22.4	53.0
18.0	171,500	46,200	44,000	26.9	25.7	48.4
16.6	156,800	42,300	44,100	27.0	28.2	44.8
16.0	152,500	41,100	44,000	26.9	28.8	44.3

SERIES I. TEST NO. 5.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 7 : 1.

Air Standard Efficiency 53.98 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 14.9°C. *Barometer* 30.29in.

Ignition Position No. 3 — 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 48 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1492	5.6	42.4	170.5	41.0	53.2	15.5	14.0	2.7	23.8	0.924	88
1484	5.7	42.3	170.0	40.7	54.4	15.5	14.0	2.7	25.4	0.872	,,
1480	5.7	42.3	170.0	40.6	55.8	15.5	14.0	2.7	27.1	0.819	,,
1500	6.0	42.0	169.0	40.9	57.6	15.6	14.0	2.7	29.4	0.748	,,
1512	6.6	41.4	167.0	40.7	61.0	15.8	14.0	2.7	33.6	0.657	,,
1512	7.7	40.3	162.5	39.6	62.6	16.0	14.0	3.0	36.8	0.618	,,
1508	9.4	38.6	156.5	38.2	63.0	16.0	14.0	3.2	39.5	0.597	,,
1522	11.0	37.0	151.5	37.0	62.3	16.1	14.2	3.5	40.8	0.594	,,
1512	12.9	35.1	144.5	35.2	60.0	16.2	14.2	3.3	43.6	0.588	,,
1496	15.1	32.9	137.0	32.9	58.3	16.3	14.3	4.0	46.2	0.592	,,
1502	5.6	5.6	—	—	59.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 7 : 1.

Lower Calorific Value of Fuel 11.330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
41.3	431,000	104,300	46,600	24.2	10.8	65.0
41.1	404,000	103,500	47,100	25.6	11.7	62.7
41.1	378,000	103,400	48,200	27.4	12.8	59.8
40.9	347,000	104,000	50,700	30.0	14.6	55.4
40.3	292,000	103,500	53,600	35.4	18.3	46.3
39.3	279,000	100,800	55,400	36.1	19.9	44.0
38.0	258,500	98,300	55,800	37.6	21.6	40.8
36.5	250,500	94,200	55,800	38.0	22.3	39.7
34.9	228,000	89,300	54,300	38.3	23.8	37.9
33.0	221,000	83,700	49,800	38.1	22.5	39.4

SERIES I. TEST NO. 6.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 7 : 1.

Air Standard Efficiency 53.98 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 14·8°C. *Barometer* 30.29in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 48 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
1284	7.6	40.4	162.0	33.5	57.8	16.5	14.3	2.5	28.8	0.935	82
1306	7.7	40.3	161.5	34.0	58.4	16.8	14.2	2.5	30.7	0.862	..
1296	7.9	40.1	160.3	33.5	61.4	17.0	14.0	2.5	37.0	0.726	..
1292	8.4	39.6	158.5	32.9	64.4	17.3	14.0	2.8	41.8	0.655	..
1296	9.3	38.7	155.0	32.4	66.6	17.5	14.0	2.8	44.8	0.620	..
1280	10.8	37.2	150.0	30.9	67.0	17.7	14.0	3.0	48.8	0.597	..
1292	13.1	34.9	142.0	29.5	65.3	17.6	14.0	3.5	52.0	0.587	..
1296	14.8	33.2	136.0	28.3	63.4	17.3	14.0	3.5	54.5	0.586	..
1302	5.05	5.05	—	—	62.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 7 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1300 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
33.9	355,000	85,400	42,000	24.0	11.8	64.2
33.9	324,000	86,600	43,000	26.8	13.3	59.9
33.7	276,000	85,400	45,200	30.9	16.4	52.7
33.1	246,000	83,900	47,000	34.1	19.2	46.7
32.5	228,000	82,600	50,700	36.2	22.3	41.5
31.4	209,000	78,800	50,700	37.7	24.2	38.1
29.7	197,000	75,200	47,800	38.1	24.3	37.6
28.4	188,000	72,200	46,400	38.2	24.7	37.1

SERIES I. TEST NO. 7.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 7 : 1.

Air Standard Efficiency 53.98 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 14.7°C. *Barometer* 30.29in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 48 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1744	5.4	42.6	175.0	49.0	54.8	16.0	14.0	3.5	26.1	0.705	92
1652	5.7	42.3	174.0	46.2	52.1	15.6	14.0	3.5	27.7	0.703	,,
1620	6.0	42.0	173.0	44.9	53.0	15.6	14.0	3.5	30.5	0.658	,,
1650	6.6	41.4	170.5	45.2	51.3	16.0	14.0	3.5	33.2	0.600	,,
1708	9.3	38.7	161.0	44.0	50.0	16.2	14.2	4.0	35.1	0.583	,,
1648	12.7	35.3	148.0	39.2	54.0	16.0	14.1	4.2	40.2	0.573	,,
1736	2.8	45.2	184.0	51.2	42.0	10.0	14.2	3.5	24.9	0.705	,,
1680	6.5	6.5	—	—	54.0	—	—	—	—	*Motoring test	

* No-heat. rich-mixture test for maximum power output only.

REMARKS.—Tests for power output and efficiency on alcohol at high speeds. Engine speed difficult to control within close limits, owing to the power output being beyond the usual range of the apparatus. A single test was made for maximum power output without any heating to carburettor, and with a fairly rich mixture. No readings of heat-loss to jacket water could be taken during these tests for reasons stated above.

SERIES I. TEST NO. 8.

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 7 : 1.

Air Standard Efficiency 53.98 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 14.6°C. *Barometer* 30.29in.

Ignition Position No. 3 = 28 deg. early.

Revs. per mn.	Torque at 3ft. rad. W = 48 lb. Spring balance	Nett weight	I.M. E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
960	9.0	39.0	154.0	23.9	54.6	17.5	14.0	2.7	41.7	0.900	72
1008	9.0	39.0	154.0	25.0	55.6	17.7	14.0	2.7	42.4	0.850	,,
970	9.1	38.9	153.7	24.2	57.4	18.0	14.0	2.7	47.1	0.790	,,
980	9.4	38.6	152.5	24.1	59.5	17.7	14.0	2.7	50.6	0.740	,,
982	9.7	38.3	151.2	23.9	61.0	18.0	14.0	2.9	52.8	0.714	,,
990*	9.8	38.2	151.0	24.1	62.7	18.0	14.0	3.0	51.9*	0.719	*Check test
990	10.5	37.5	149.0	23.7	64.8	18.2	14.2	3.3	59.6	0.637	72
978	11.7	36.3	144.5	22.7	65.9	18.6	14.2	3.5	63.9	0.620	,,
964	13.4	34.6	139.0	21.9	65.0	18.5	14.0	3.7	66.8	0.613	,,
980	14.3	33.7	135.5	21.4	64.4	18.3	13.8	4.0	69.8	0.602	,,
986	15.6	32.4	130.5	20.7	61.7	18.5	13.8	4.2	73.2	0.596	,,
977	4.35	4.35	—	—	63.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 7 : 1.

Lower Calorific Value of Fuel 11.330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 975 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
24.3	244,000	61,000	33,600	25.0	13.8	61.2
24.3	241,000	63,700	34,000	26.5	14.2	59.3
24.25	217,500	61,700	36,000	28.4	16.6	55.0
24.05	202,000	61,400	37,500	30.4	18.5	51.1
23.95	193,500	60,800	38,500	31.4	19.9	49.7
23.85	196,500	61,400	39,100	31.3	19.9	48.8
23.55	171,500	60,400	40,000	35.2	23.3	41.5
22.8	160,000	57,900	40,200	36.2	25.1	38.7
22.0	152,500	55,800	40,400	36.6	26.4	37.0
21.4	146,000	54,600	39,700	37.4	27.2	35.4
20.6	140,500	52,700	38,200	37.6	27.2	35.2

SERIES I. TEST CHECK "*a*."

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 14.8°C. *Barometer* 30.2in.

Ignition Position No. 3.2 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 32 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature °C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
1508	3.2	28.8	119.6	29.0	61.35	25.2	14.0	3.5	55.5	0.57	87.5
1512	4.0	28.0	116.8	28.4	61.7	25.0	14.0	3.7	58.3	0.543	
1504	5.6	26.4	111.0	26.9	60.7	25.5	14.2	3.9	62.0	0.540	
1516	6.1	25.9	109.5	26.6	59.8	26.0	14.2	4.3	62.8	0.537	
1514	6.6	25.4	107.5	26.2	59.8	26.0	14.2	4.5	64.0	0.537	
1508	8.0	24.0	102.5	24.8	58.6	26.5	14.2	4.5	65.4	0.554	
1496	9.4	22.6	97.7	23.5	57.25	26.8	14.2	5.0	68.6	0.560	
1500	4.9	4.9	17.4	—	60.0	—	—	—	—	Motoring test	

REMARKS.—Tests for efficiency and jacket losses on petrol at varying speeds with a compression-ratio of 3.8 : 1 for comparison with similar tests on ethyl alcohol.

HEAT-BALANCE SHEET.

Check Test on Petrol "A." *Compression-Ratio* 3.8 : 1.

Lower Calorific Value of Fuel 18,200 B.Th.Us. per pint at 15°C.

I.H.P. (correeted to 1500 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
28.9	302,000	74,000	75,100	24.6	24.9	50.5
28.2	283,500	72,400	75,500	25.8	26.6	47.6
26.8	265,000	68,600	74,800	25.9	28.2	45.9
26.3	259,000	67,800	72,400	26.1	28.0	45.9
25.9	253,000	66,800	72,400	26.1	28.6	45.3
24.6	250,000	63,300	72,200	25.3	28.9	45.8
23.6	240,000	59,900	69,000	25.0	28.8	46.2

SERIES I. TEST CHECK "*b*."

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 14·8°C. *Barometer* 30.2in.

Ignition Position No. 3.2 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 32 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
1676	4.8	27.2	115.5	31.1	62.2	26.0	14.3	4.5	54.7	0.532	92
1674	4.8	27.2	115.5	31.1	64.27	25.0	14.5	4.5	54.9	0.530	
1668	2.8	29.2	122.5	32.8	66.75	25.5	14.4	4.5	51.4	0.535	
1680	5.3	5.3	18.9	—	66.0	—	—	—	—	Motoring test	

REMARKS.—Tests for efficiency and jacket losses on petrol at varying speeds with a compression-ratio of 3.8 : 1 for comparison with similar tests on ethyl alcohol.

HEAT-BALANCE SHEET.

Check Test on Petrol "A." *Compression-Ratio* 3.8 : 1.

Lower Calorific Value of Fuel 18,200 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1675 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
31.1	301,500	79,300	75,500	26.3	25.2	48.5
31.1	299,500	79,300	77,000	26.4	25.7	47.8
32.9	321,000	83,900	81,000	26.2	25.3	48.5

SERIES I. TEST CHECK "C."

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 14.8°C *Barometer* 30.2in

Ignition Position No. 3.2 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 32 lb. Spring balance	Nett weight	I.M. E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Heat input
											Volts
972	3.9	28.1	113.5	17.7	61.15	25.0	14.0	3.0	80.7	0.63	72
972	4.1	27.9	112.5	17.6	62.3	25.5	13.5	3.2	87.7	0.584	
976	4.6	27.4	110.9	17.4	63.9	25.9	14.2	3.2	93.9	0.55	
976	5.2	26.8	108.7	17.03	64.35	26.5	13.8	3.3	97.9	0.54	
976	6.7	25.3	103.5	16.2	63.55	26.7	14.0	3.3	102.4	0.54	
970	8.8	23.2	95.9	15.0	62.8	26.9	13.9	3.5	109.8	0.55	
975	3.8	3.8	14.5	—	62.0	—	—	—	—	Motoring test	

REMARKS.—Tests for efficiency and jacket losses on petrol at varying speeds with a compression-ratio of 3.8 : 1 for comparison with similar tests on ethyl alcohol.

HEAT-BALANCE SHEET.

Check Test on Petrol "A." *Compression-Ratio* 3.8 : 1.

Lower Calorific Value of Fuel 18,200 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 975 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
17.75	203,000	45,200	48,300	22.2	23.8	54.0
17.65	187,500	44,900	49,000	24.0	26.2	49.8
17.4	174,100	44,400	50,900	25.5	29.2	45.3
17.03	168,000	43,400	51,000	25.9	30.4	43.7
16.2	160,000	41,300	49,500	25.9	30.9	43.2
15.1	150,500	38,250	48,700	25.5	32.4	42.1

SERIES I. TEST CHECK "*d*."

Effect of Piston Speed on Power Output and Efficiency.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 13°C. *Barometer* 30.07in.

Ignition Position No. 3.2 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Intake heater
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1488	6.8	33.2	137.3	33.0	58.0	25.5	14.0	3.2	45.0	0.606	88.0
1500	6.5	33.5	138.3	33.5	60.0	26.0	14.0	3.2	52.6	0.511	88.0
1496	6.7	33.3	137.7	33.2	61.0	27.0	14.0	3.2	55.2	0.491	88.0
1505	7.2	32.8	135.9	33.0	69.0	27.4	14.0	3.5	59.0	0.462	87.8
1504	7.15	32.85	136.1	33.1	60.0	27.8	14.1	3.5	59.1	0.461	87.8
1501	7.9	32.1	133.4	32.3	59.0	28.0	14.3	3.9	62.2	0.448	87.7
1503	8.65	31.35	130.7	31.7	58.0	27.3	14.2	4.4	64.0	0.443	87.3
1504	9.3	30.7	128.5	31.2	57.0	27.8	14.5	4.8	65.7	0.439	88.0
1507	10.5	29.5	124.3	30.2	63.0	27.9	14.8	5.0	68.3	0.436	87.2
1502	11.2	28.8	121.8	29.5	63.0	28.0	15.0	5.4	70.8	0.431	87.5
1500	12.7	27.3	116.4	28.2	61.0	28.4	15.0	5.5	73.8	0.433	88.0
1501	12.5	27.5	117.1	28.4	61.0	28.8	15.0	5.8	73.7	0.431	88.3
1500*	16.0	24.0	104.8	25.35	61.0	28.0	15.0	3.2	73.7	0.482	—
1486	6.3	33.7	139.0	33.4	60.0	27.0	15.0	3.2	53.4	0.505	87.8
1513	6.3	33.7	139.0	33.9	60.0	26.8	15.1	3.2	49.5	0.537	88.0
1500	5.6	5.6	19.8	—	60.0	—	—	—	—	Motoring test	

* Slightly unstable.

SERIES II. TEST NO. 1.

Throttling to 1500 Revs. per minute, 100 per cent Torque.

Compression-Ratio 5.45 : 1.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 17°C. *Barometer* 30.35in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb. Spring balance	Nett weight	I.M. E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Air intake heater
											Volts
1513	1.0	39.0	156.6	38.5	60.0	16.5	18.0	3.2	28.4	0.825	88
1492	1.15	38.85	156.0	37.8	59.0	16.5	18.0	3.2	28.0	0.850	,,
1513	1.00	39.0	156.6	38.6	57.5	16.3	18.0	3.2	28.4	0.821	,,
1503	1.3	38.7	155.7	38.1	58.0	17.0	18.0	3.2	30.8	0.768	,,
1538	1.3	38.7	155.7	38.9	57.5	17.0	18.5	3.2	30.9	0.750	,,
1505	1.8	38.2	153.8	37.6	56.6	17.0	18.5	3.5	33.8	0.708	,,
1494	3.3	36.7	148.6	36.1	59.3	17.0	18.5	3.5	37.4	0.667	,,
1519	4.7	35.3	143.7	35.5	58.8	17.0	18.5	3.7	38.4	0.660	,,
1510	6.5	33.5	137.3	33.7	57.9	17.0	18.8	3.8	40.4	0.661	,,
1506	7.9	32.1	132.5	32.4	55.5	17.0	19.0	4.0	43.0	0.646	,,
1516	9.3	30.7	127.5	31.4	53.9	16.8	18.8	4.1	44.0	0.652	,,
1496	12.2	27.8	117.4	28.6	52.6	16.8	18.6	4.3	46.4	0.678	,,
1500	5.6	5.6	20.0	—	60.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
38.0	330,000	99,000	62,000	30.0	18.8	51.2
37.5	302,000	95,800	64,000	31.7	21.2	47.1
36.2	272,500	92,000	63,800	33.8	23.4	42.8
35.1	265,000	90,400	62,800	34.1	23.7	42.2
33.4	252,500	85,800	60,200	34.1	23.9	42.0
32.2	237,000	82,500	56,500	34.8	23.9	41.3
31.1	231,500	80,000	55,800	34.6	24.1	41.3
28.7	220,000	72,800	52,700	33.1	23.9	43.0

SERIES II. TEST NO. 2.

Throttling to 1500 Revs. per minute, 80 per cent Torque.

Compression-Ratio 5.45 : 1.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 18°C. *Barometer* 30.35in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M.E.P.	Ind. H.P.	Temperature ° C.			Ingition position No.	Fuel consumption. Unit vol. ¼ pint.		Air intake heater
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1501	8.8	31.2	130.0	31.8	57.5	16.6	18.5	3.6	35.0	0.808	78
1505	8.9	31.1	129.5	31.7	59.0	17.2	18.5	3.6	38.4	0.740	,,
1504	9.2	30.8	128.5	31.5	60.2	17.3	18.2	3.7	39.4	0.726	,,
1512	10.0	30.0	125.7	30.9	60.1	17.7	18.2	3.8	43.3	0.673	,,
1502	11.3	28.7	121.1	29.6	58.4	18.0	18.5	4.0	46.2	0.658	,,
1508	13.1	26.9	114.8	28.1	56.5	17.7	19.0	4.2	49.6	0.646	,,
1482	14.5	25.5	110.0	26.5	53.9	18.0	20.0	4.4	53.1	0.640	,,
1500	16.4	23.6	103.2	25.2	52.6	17.7	22.0	4.6	55.0	0.650	,,
1496	19.8	20.2	91.4	22.2	51.15	17.5	20.0	4.8	57.7	0.703	,,
1500	5.8	5.8	20.6	—	61.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
31.4	259,000	80,200	58,500	31.0	22.6	46.4
30.6	235,500	78,700	57,800	33.4	24.5	42.1
29.5	220,500	75,300	55,800	34.1	25.3	40.6
28.0	205,500	71,600	53,400	34.9	26.0	39.1
26.8	192,000	67,500	50,300	35.1	26.2	38.7
25.2	185,500	64,200	48,200	34.6	26.0	39.4
22.3	176,500	56,500	46,500	32.0	25.8	42.2

SERIES II. TEST No. 3.

Throttling to 1500 Revs. per minute, 60 per cent Torque.

Compression-Ratio 5.45 : 1.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 18°C. *Barometer* 30.35in.

Ignition Position No. 3 — 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 24 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ½ pint. Seconds	Pint per I.H.P. unit vol. per hr.	Air intake heater
											Volts
1510	1.4	22.6	100.2	24.6	61.5	18.0	19.0	4.0	50.5	0.725	68
1494	2.2	21.8	97.4	23.6	61.2	18.0	18.5	4.0	57.9	0.659	..
1500	4.0	20.0	91.1	22.2	59.1	18.0	19.0	4.3	62.2	0.652	..
1496	5.7	18.3	85.2	20.7	56.2	18.5	21.0	4.7	68.0	0.640	..
1493	7.6	16.4	78.5	19.0	53.5	18.0	22.0	4.9	72.2	0.656	..
1507*	9.5	14.5	71.8	17.55	51.7	17.3	21.5	5.0	74.0	0.694	..
1500	5.9	5.9	21.0	—	60.0	—	—	—	—	Motoring test	

* Unstable.

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
23.7	176,000	60,100	49,700	34.1	28.2	37.7
22.2	164,000	56,500	47,400	34.5	28.9	36.6
20.8	150,000	52,700	44,000	35.1	29.3	35.6
19.1	141,200	48,400	41,300	34.3	29.2	36.5
17.5	137,800	44,700	39,300	32.5	28.5	39.0

SERIES II. TEST NO. 4.

Throttling to 1500 Revs. per minute, 40 per cent Torque.

Compression-Ratio 5.45 : 1.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 19°C. *Barometer* 30.35in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 24 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ½ pint.		Air intake heater
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1500	8.8	15.2	74.5	18.2	58.7	18.0	18.0	4.0	64.0	0.773	56
1498	8.9	15.1	74.2	18.0	58.6	18.0	18.0	4.0	65.9	0.760	,,
1497	10.4	13.6	68.9	16.75	58.6	17.9	19.0	4.6	83.0	0.648	,,
1503	11.5	12.5	65.1	15.9	55.5	17.5	20.2	5.0	88.0	0.643	55
1501	12.8	11.2	60.5	14.75	54.5	16.8	18.2	5.3	93.2	0.655	50
1494*	15.8	8.2	50.0	12.1	53.5	17.9	18.0	5.8	98.5	0.755	,,
1500	6.0	6.0	21.3	—	62.0	—	—	—	—	Motoring test	

* Popped twice.

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
18.0	155,000	45,800	40,500	29.5	26.1	44.4
17.0	122,900	42,700	40,000	34.7	32.5	32.8
15.9	115,800	40,500	38,000	35.0	32.8	32.2
14.7	109,300	37,500	36,200	34.3	33.1	32.6
12.1	103,500	30,800	34,200	29.8	33.0	37.2

SERIES II. TEST No. 5.

Throttling to 950 Revs. per minute, 100 per cent Torque.
Compression-Ratio 5.45 : 1.
Air Standard Efficiency 49.4 per cent.
Fuel—**Ethyl Alcohol 95 Vol. per cent.**
S.G. 0.815. *Mean Temp.* 18°C. *Barometer* 30.28in.
Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb. Spring balance	Torque at 3ft. rad. W = 40 lb. Nett weight	I.M.E.P.	Ind. H.P.	Temperature °C. Water outlet	Temperature °C. Inlet housing	Temperature °C. Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Fuel consumption. Unit vol. ¼ pint. Pint per I.H.P. per hr.	Air intake heater
											Volts
918	4.6	35.4	139.8	20.9	53.5	18.0	16.3	2.9	41.4	1.04	68
926	4.4	35.6	140.6	21.2	53.5	17.5	16.5	3.0	38.7	1.10	,,
926	4.5	35.5	140.3	21.1	53.8	17.6	16.2	3.0	41.2	1.035	,,
944	4.7	35.3	139.5	21.4	55.2	17.6	16.2	3.0	43.0	0.980	,,
937	4.8	35.2	139.2	21.2	56.9	18.0	16.5	3.0	45.3	0.938	,,
936	4.9	35.1	138.8	21.1	59.3	18.5	16.5	3.0	51.3	0.832	,,
935	5.1	34.9	138.1	21.0	60.9	18.5	16.6	3.0	55.6	0.771	,,
936	5.8	34.2	135.6	20.65	62.5	18.8	16.7	3.2	59.3	0.735	,,
934	6.7	33.3	132.5	20.1	63.2	18.6	16.8	3.2	64.4	0.695	,,
926	7.7	32.3	129.0	19.4	63.0	18.8	17.0	3.5	67.2	0.691	,,
932	9.0	31.0	124.5	18.85	62.0	18.8	17.0	3.6	69.5	0.688	,,
934	11.1	28.9	117.0	17.75	59.8	18.6	17.0	3.9	74.7	0.680	,,
936	12.4	27.6	112.4	17.1	57.0	18.5	17.0	4.0	76.0	0.693	,,
940	4.4	4.4	15.6	—	58.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1.
Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 950 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	B.Th.Us. per hour. Heat to I.H.P.	B.Th.Us. per hour. Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
21.6	248,000	53,800	34,000	21.7	13.7	64.6
21.5	237,000	54,500	35,300	22.9	14.9	62.2
21.5	225,000	54,000	36,700	23.9	16.3	59.8
21.4	198,000	53,800	38,300	27.0	19.3	53.7
21.3	183,500	53,500	40,000	29.1	21.8	49.1
21.0	172,000	52,800	41,200	30.5	23.9	45.6
20.4	158,500	51,200	41,700	32.3	26.3	41.4
19.9	152,000	49,400	41,300	32.5	27.1	40.4
19.2	147,000	48,000	40,300	32.7	27.4	39.9
18.0	136,500	45,100	38,700	33.0	28.3	38.7
17.3	134,500	43,500	36,000	32.4	26.8	40.8

SERIES II. TEST No. 6.

Throttling to 950 Revs. per minute, 80 per cent Torque.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 18.5°C. *Barometer* 30.28in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 32 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature °C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Air intake heater
											Volts
936	3.9	28.1	115.5	17.6	58.0	18.5	16.6	3.2	67.1	0.763	61
956	4.4	27.6	113.8	17.7	56.9	18.3	16.6	3.3	72.7	0.700	,,
944	5.7	26.3	109.2	16.8	52.5	18.0	16.6	3.5	78.0	0.687	,,
936	7.2	24.8	104.0	15.85	61.0	18.3	16.8	3.8	83.5	0.680	,,
936	8.3	23.7	101.1	15.25	63.0	18.2	16.9	4.0	88.4	0.668	,,
952	9.5	22.5	95.9	14.85	61.9	18.0	17.0	4.2	91.8	0.661	,,
936	12.7	19.3	84.7	12.9	59.6	18.0	17.0	4.6	98.5	0.708	,,
950	4.8	4.8	17.0	—	58.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 11,339 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 950 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
17.6	140,000	45,100	32,300	32.2	23.1	44.7
16.9	131,000	42,800	32,900	32.7	25.1	42.2
16.1	122,000	40,300	34,300	33.0	28.1	38.9
15.5	115,500	38,800	33,200	33.6	28.7	37.7
14.8	111,000	37,800	32,000	33.9	28.8	37.3
13.1	103,500	32,900	29,500	31.8	28.5	39.7

SERIES II. TEST NO. 7.

Throttling to 950 Revs. per minute, 60 per cent Torque.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 18.5°C. *Barometer* 30.28in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 24 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Air intake heater
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
935	2.8	21.2	91.8	13.95	64.0	19.0	17.0	4.0	90.0	0.717	46.5
940	4.0	20.0	87.6	13.4	63.9	18.5	17.0	4.3	99.0	0.678	,,
932	5.4	18.6	82.7	12.5	61.2	17.7	17.2	4.4	107.3	0.670	,,
936	7.2	16.8	76.4	11.6	58.0	17.2	17.5	5.0	114.6	0.677	,,
945	4.9	4.9	17.4	—	57.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 950 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
13.5	103,000	34,100	32,800	33.1	31.8	35.1
12.7	95,000	31,800	30,700	33.5	32.3	34.2
11.7	89,000	29,500	28,700	33.1	32.2	34.7

SERIES II. TEST NO. 8.

Throttling to 950 Revs. per minute, 40 per cent Torque.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 19°C. *Barometer* 30.28in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 16 lb. Spring balance	Nett weight	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ½ pint. Seconds unit vol.	Pint per I.H.P. per hr.	Air intake heater
											Volts
932	1.8	14.2	67.4	10.2	67.5	20.0	17.6	4.0	107.3	0.822	43
944	1.8	14.2	67.4	10.3	68.3	20.3	17.6	4.2	111.6	0.783	,,
944	2.2	13.8	66.0	10.1	67.9	20.0	17.2	4.5	132.4	0.673	,,
944	3.4	12.6	61.7	9.4	65.5	19.6	17.2	5.0	143.0	0.670	,,
942	4.2	11.8	59.0	8.95	62.5	19.0	17.2	5.0	149.4	0.672	,,
939	5.9	10.1	53.0	8.0	60.6	18.6	17.2	5.3	157.6	0.714	,,
950	4.9	4.9	17.6	—	59.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Ethyl Alcohol. *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 11,330 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 950 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
10.4	91,500	26,200	26,300	28.6	28.7	42.7
10.2	77,000	25,700	27,000	33.4	35.0	31.6
9.45	71,490	23,900	25,400	33.5	35.6	30.9
9.05	68,300	22,800	24,100	33.4	35.3	31.3
8.1	64,800	20,400	22,600	31.5	34.9	33.6

SERIES II. TEST CHECK "*a*."

Effect of Varying Throttle Opening—Full Throttle.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 17°C. *Barometer* 30.4in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input to carburettor
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1502	5.5	34.5	141.5	34.5	59.0	22.5	18.0	3.2	39.7	0.656	88
1504	5.4	34.6	141.9	34.7	61.9	25.3	18.0	3.2	45.8	0.565	,,
1508	5.5	34.5	141.5	34.6	67.3	26.8	18.8	3.2	54.4	0.476	,,
1506	5.8	34.2	139.5	34.1	69.8	27.8	19.5	3.3	59.3	0.443	,,
1516	7.5	32.5	133.8	33.0	68.6	29.5	20.0	3.5	63.6	0.427	,,
1504	10.4	29.6	123.5	30.3	64.3	29.6	20.5	3.9	70.8	0.418	,,
1512	11.0	29.0	121.0	29.9	61.0	29.2	20.0	3.9	71.8	0.419	,,
1498	13.4	26.6	112.5	27.6	59.5	29.2	19.5	4.2	76.5	0.426	,,
1500	5.6	5.6	—	—	62.0	—	—	—	—	Motoring test	

HEAT-BALANCE SHEET.

Check Test on Petrol "A." *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 18,200 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
34.5	416,000	88,000	52,700	21.1	12.7	66.2
34.55	355,000	88,300	55,800	24.9	15.7	59.4
34.4	300,000	88,000	61,200	29.3	20.4	50.3
34.1	277,000	87,300	64,000	31.6	23.1	45.3
32.3	253,000	83,000	63,200	32.8	25.0	42.2
29.5	226,000	75,600	58,700	33.7	26.0	40.3
28.9	220,000	74,000	55,800	33.7	25.4	40.7
26.6	206,500	67,900	53,000	32.8	25.6	41.6

100 per cent full-load torque (full throttle).

SERIES II. TEST CHECK "*b*."

Effect of Varying Throttle Opening—80 per cent Torque.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 17°C. *Barometer* 30.4in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 32 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ½ pint.		Heat input to carburettor
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1508	3.7	28.3	119.5	29.5	65.0	34.0	19.5	3.4	47.4	0.643	78
1508	3.1	28.9	121.5	30.0	68.2	34.2	19.5	3.5	54.6	0.548	,,
1524	2.9	29.1	122.0	30.4	74.0	35.5	20.0	3.6	64.7	0.455	,,
1500	4.2	27.8	117.7	28.9	75.0	37.5	20.0	3.8	71.3	0.434	,,
1498	6.4	25.6	113.6	27.7	72.1	38.2	20.5	4.0	79.7	0.408	,,
1504	8.1	23.9	103.8	25.5	67.1	38.7	20.0	4.3	84.4	0.417	,,
1500	5.8	5.8	—	—	65.0	—	—	—	—	Motoring test	

REMARKS.—Test throttled down to 80 per cent full-load torque. Motoring test with throttle locked in position giving 80 per cent torque.

HEAT-BALANCE SHEET.

Check Test on Petrol "A." *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 18,200 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
28.2	332,000	72,200	48,500	21.8	14.6	63.6
28.8	288,000	73,700	51,200	25.6	17.7	56.7
28.9	241,000	74,200	54,500	30.8	22.6	46.6
27.8	219,000	70,900	55,000	32.4	25.1	42.5
25.6	190,000	65,200	54,300	34.3	28.6	37.1
23.9	181,000	61,000	49,200	33.7	27.2	39.1
	80 per cent full-load torque.					

SERIES II. TEST CHECK "c."

Effect of Varying Throttle Opening—60 per cent Torque.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Petrol " A."**

S.G. 0.782. *Mean Temp.* 17°C. *Barometer* 30.4in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 24 lb. Spring balance	Torque at 3ft. rad. W = 24 lb. Nett weight	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Temperature ° C. Inlet housing	Temperature ° C. Engine room	Ignition position No.	Fuel consumption. Unit vol. ½ pint. Seconds unit vol.	Fuel consumption. Unit vol. ½ pint. Pint per I.H.P. per hr.	Heat input to carburettor
											Volts
1504	3.3	20.7	93.5	22.9	68.7	37.0	19.5	3.7	64.2	0.640	68
1504	3.1	20.9	93.9	23.1	70.5	39.2	19.5	3.8	74.0	0.525	,,
1504	3.0	21.0	94.2	23.2	75.8	40.5	20.0	4.0	86.2	0.455	,,
1498	3.8	20.2	91.3	22.4	78.0	43.0	20.5	4.2	95.0	0.422	,,
1502	5.1	18.9	86.8	21.3	77.2	45.0	21.0	4.5	101.6	0.412	,,
1500	6.9	17.1	80.5	19.8	72.3	45.5	20.5	4.7	108.3	0.418	,,
1500	9.6	14.4	71.0	17.4	67.4	45.0	20.0	5.0	115.6	0.447	,,
1500	5.9	5.9	—	—	68.0	—	—	—	—	Motoring test	

REMARKS.—Test throttled down to 60 per cent full-load torque. Motoring test with throttle locked in position giving 60 per cent torque.

HEAT-BALANCE SHEET.

Check Test on Petrol " A." *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 18,200 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour. Total heat of fuel supplied	B.Th.Us. per hour. Heat to I.H.P.	B.Th.Us. per hour. Heat to cooling water	Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
22.9	266,000	58,400	46,000	22.0	17.3	60.7
23.1	212,000	58,900	46,500	27.8	22.0	50.2
23.2	193,000	59,200	46,700	30.7	24.2	45.1
22.4	172,000	57,100	47,800	33.2	27.8	39.0
21.3	159,000	54,300	48,400	34.2	30.4	35.4
19.8	151,000	50,500	48,100	33.4	31.9	34.7
17.4	141,000	44,400	44,000	31.5	31.2	37.3
	60 per cent full-load torque.					

SERIES II. TEST CHECK "*d*.'

Effect of Varying Throttle Opening—40 per cent Torque.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 17°C. *Barometer* 30.4in.

Ignition Position No. 3 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 24 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ½ pint.		Heat input to carburettor
	Spring balance	Nett weight			Water outlet	Inlet housing	Engine room		Seconds unit vol.	Pint per I.H.P. per hr.	
											Volts
1510	10.2	13.8	69.4	17.1	64.0	37.0	19.0	4.0	89.7	0.585	56
1500	10.0	14.0	70.0	17.1	61.8	37.5	19.5	4.0	94.3	0.555	,,
1520	10.2	13.8	69.4	17.2	62.0	39.5	19.5	4.5	100.8	0.518	,,
1500	10.5	13.5	68.3	16.7	51.0	42.0	19.5	4.5	125.7	0.430	,,
1500	11.0	13.0	66.6	16.3	62.0	44.0	20.0	4.7	130.8	0.421	,,
1500	12.4	11.6	61.6	15.1	59.7	46.0	20.0	5.0	139.2	0.429	,,
1496	14.6	9.4	54.0	13.2	58.6	46.0	20.0	5.0	146.8	0.451	,,
1500	6.0	6.0	—	—	59.0	—	—	—	—	Motoring test	

REMARKS.—Test throttled down to 40 per cent full-load torque. Motoring test with throttle locked in position, giving 40 per cent torque.

HEAT-BALANCE SHEET.

Check Test on Petrol "A." *Compression-Ratio* 5.45 : 1.

Lower Calorific Value of Fuel 18,200 B.Th.Us. per pint at 15°C.

I.H.P. (corrected to 1500 r.p.m.)	B.Th.Us. per hour.			Heat to I.H.P. Per cent.	Heat to cooling water Per cent.	Heat to exhaust radiation, etc. Per cent.
	Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water			
17.0	182,000	43,600	40,700	23.9	22.4	53.7
17.1	173,000	43,600	40,700	25.2	23.6	51.2
17.0	162,000	43,800	40,800	27.1	25.2	47.7
16.7	131,500	42,600	—	32.4	—	—
16.3	124,500	41,600	40,800	33.2	32.7	33.9
15.1	118,000	38,500	39,400	32.7	33.4	33.9
13.2	108,500	33,700	38,400	31.1	35.3	33.6
	40 per cent. full-load torque.					

SERIES III. TEST NO. 1.

Effect of Cold Cylinder.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 14°C. *Barometer* 29.75in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C		Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heater	Air temp.
	Spring balance	Nett weight			Water outlet	Inlet housing		Seconds unit vol.	Pint per I.H.P. per hr.		
										Volts	° C.
1510	2.1	37.9	156.8	38.0	34.0	13.7	3.2	28.6	0.829	86.5	
1502	2.2	37.8	156.5	37.6	34.0	14.0	3.5	30.5	0.785	88.0	14.0
1510	2.6	37.4	155.0	37.5	34.2	14.0	3.2	32.1	0.747	87.5	
1500	3.9	36.1	150.5	36.1	34.0	14.0	4.2	35.2	0.708	88.0	
1498	4.7	35.3	147.6	35.4	33.0	14.5	4.5	36.3	0.701	87.8	14.1
1518	5.7	34.3	144.1	35.0	33.0	14.3	4.5	37.4	0.688	87.5	
1510	6.6	33.4	141.0	34.0	33.0	14.1	5.0	38.4	0.689	87.5	
1491	7.6	32.4	137.0	32.7	33.0	14.0	5.5	40.0	0.689	87.0	14.0
1510*	11.5	28.5	123.4	29.9	32.0	13.5	4.5	41.9	0.720	88.0	
1505	2.0	38.0	157.1	37.9	33.0	13.5	3.2	25.5	0.932	87.0	
1500	2.1	37.9	156.8	37.7	33.0	13.5	3.2	28.0	0.853	86.5	
1520	2.0	38.0	157.0	38.1	32.0	13.8	3.2	27.1	0.872	87.5	14.0

* Slightly unstable.

REMARKS.—Motoring test for friction :—Hot : 5.3 lb., 1505 r.p.m. at 98°C. water jacket temperature ; cold : 6.1 lb., 1500 r.p.m. at 32°C. water jacket temperature. ∴ Friction hot is 19 lb. M.E.P.; friction cold is 21.5 lb. M.E.P.

SERIES III. TEST No. 2.

Effect of Hot Cylinder.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 14°C. *Barometer* 29.75in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.		Ignition position No.	Fuel consumption. Unit vol. ½ pint.		Heater	Air temp.
	Spring balance	Nett weight			Water outlet	Inlet housing		Seconds unit vol.	Pint per I.H.P. per hr.		
										Volts	° C.
1503	3.7	36.3	148.7	35.7	82.0	14.0	3.2	28.4	0.864	87.0	14.5
1510	4.2	35.8	146.9	35.4	86.0	14.4	3.2	31.6	0.805	88.0	
1495	4.7	35.3	145.1	34.7	90.0	14.8	3.2	35.0	0.741	87.0	
1507	5.9	34.1	140.8	33.9	90.0	15.2	3.5	37.1	0.717	86.5	
1510	6.6	33.4	138.3	33.4	89.0	15.0	3.7	38.0	0.709	87.5	
1509	8.0	32.0	133.3	32.2	88.0	15.2	3.8	39.6	0.706	87.5	
1517	8.8	31.2	130.5	31.6	87.0	15.0	3.8	40.8	0.698	88.0	
1520	10.0	30.0	126.1	30.6	87.0	15.2	4.1	42.4	0.694	86.5	14.0
1500	11.5	28.5	120.9	29.0	86.5	15.0	4.5	44.5	0.697	87.5	
1508	13.0	27.0	115.4	27.8	85.0	14.8	4.8	45.6	0.710	87.0	
1500	14.7	25.3	109.3	26.2	83.0	14.6	4.8	46.7	0.736	86.5	13.5
1504	3.8	36.2	148.3	35.7	89.0	15.5	3.2	27.6	0.913	87.5	

SERIES IV. TEST NO. 1.

Effect of No Heat to Intake.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 14°C. *Barometer* 30.09in.

Ignition Position No. 3.2 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb. Spring balance	Torque at 3ft. rad. W = 40 lb. Nett weight	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Temperature ° C. Inlet housing	Ignition position No.	Fuel consumption. Unit vol. ½ pint. Seconds unit vol.	Fuel consumption. Unit vol. ½ pint. Pint per I.H.P. per hr.	Heater	Room air temp.
										Volts	° C.
1515	1.1	38.9	157.4	38.5	62.0	7.0	3.2	27.4	0.854	Nil	13.8
1497	1.1	38.9	157.4	38.0	58.0	7.0	,,	28.7	0.826	,,	
1509	1.35	38.65	156.6	38.1	59.0	7.0	,,	30.7	0.770	,,	13.9
1501	2.1	37.9	153.9	37.3	59.5	7.5	3.5	33.4	0.723	,,	14.0
1490	2.6	37.4	152.1	36.6	60.0	7.5	3.8	34.5	0.713	,,	
1508	3.0	37.0	150.6	36.7	60.0	7.6	,,	34.4	0.713	,,	
1495	3.0	37.0	150.6	36.4	59.0	7.6	,,	34.7	0.713	,,	
1505	3.6	36.4	148.5	36.1	60.0	8.0	4.0	35.4	0.704	,,	14.0
1498	4.1	35.9	146.8	35.5	59.0	8.0	4.3	36.2	0.701	,,	
1512	4.6	35.4	145.0	35.4	59.0	8.0	4.5	36.2	0.703	,,	
1491	5.3	34.7	142.5	34.4	60.0	8.1	5.0	37.5	0.698	,,	14.0
1508	5.65	34.35	141.2	34.4	60.0	8.1	5.5	37.7	0.694	,,	
1506	6.6	33.4	138.0	33.55	59.0	8.0	5.5	38.7	0.694	,,	
1507	7.0	33.0	136.6	33.2	59.0	8.0	5.9	39.0	0.695	,,	14.2
1498	6.8	33.2	137.2	33.2	59.0	8.0	5.5	39.1	0.694	,,	
1517	6.8	33.2	137.2	33.6	59.0	8.0	5.0	38.6	0.694	,,	
1501*	8.7	31.3	130.5	31.7	59.0	8.0	5.8	40.1	0.708		
1508*	7.8	32.2	133.8	32.5	58.0	7.9	5.8	39.5	0.702		
1495	0.9	39.1	158.0	38.2	60.0	7.8	3.2	28.9	0.816	Nil	

* Slightly unstable.

SERIES IV. TEST NO. 2.

Effect of Heating Intake.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.815. *Mean Temp.* 15°C. *Barometer* 30.07in.

Ignition Position No. 3.2 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.		Ignition position No.	Fuel consumption. Unit vol. ½ pint.		Heater	Room air temp.
	Spring balance	Nett weight			Water outlet	Inlet housing		Seconds unit vol.	Pint per I.H.P. per hr.		
										Volts	° C.
1512	4.25	35.75	146.3	35.7	59.0	18.9	3.2	30.0	0.840	108.3	15.8
1504	4.0	36.0	147.3	35.8	56.0	19.0	,,	25.7	0.978	108.0	15.8
1500	4.2	35.8	146.5	35.4	60.0		,,	28.0	0.908	108.0	
1490	4.5	35.5	145.4	35.0	61.0	19.5	,,	33.2	0.775	107.8	16.0
1503	4.9	35.1	144.1	34.9	62.0		,,	34.5	0.749	107.5	
1498	5.7	34.3	141.1	34.1	59.0	19.3	3.5	37.0	0.713	108.0	16.1
1503	6.75	33.25	137.5	33.35	59.0		3.8	38.8	0.696	107.5	
1507	7.3	32.7	135.5	33.0	62.0	19.2	4.0	39.5	0.691	107.5	16.0
1515	7.7	32.3	134.0	32.8	55.0		4.6	39.6	0.693	107.0	
1508	7.4	32.6	135.2	32.9	58.0	19.3	4.6	39.8	0.688	107.0	16.0
1508	8.8	31.2	130.3	31.7	59.0		4.6	41.7	0.681	107.5	
1509	9.6	30.4	127.4	31.0	59.0	19.3	5.0	42.6	0.681	108.0	16.0
1512	10.3	29.7	125.0	30.5	60.0		5.2	43.8	0.675	108.0	
1497	11.25	28.75	121.6	29.35	61.0		5.5	45.4	0.676	107.0	
1503	12.0	28.0	118.9	28.9	60.0	19.0	5.8	46.3	0.673	106.0	16.0
1505*	13.4	26.6	114.0	27.7	60.0		5.9	47.2	0.690	105.0	
1495	4.3	35.7	146.1	35.3	64.0	19.3	3.2	31.3	0.815	108.0	15.9
1489	4.1	35.9	147.0	35.3	62.0		3.2	27.4	0.931	108.0	

* Slightly unstable.

REMARKS.—Heater to intake : 108 volts = 100 B.Th.Us. per minute (energy output varies as the voltage squared).

SERIES III. AND IV. TEST CHECK "*a*."

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Petrol "A."**

S.G. 0.782. *Mean Temp.* 13°C. *Barometer* 30.07in.

Ignition Position No. 3.2 = 28 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.		Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heater	Room air temp.
	Spring balance	Nett weight			Water outlet	Inlet housing		Seconds unit vol.	Pint per I.H.P. per hr.		
										Volts	° C.
1488	6.8	33.2	137.3	33.0	58.0	25.5	3.2	45.0	0.606	88.0	14.0
1500	6.5	33.5	138.3	33.5	60.0	26.0	,,	52.6	0.511	88.0	14.0
1496	6.7	33.3	137.7	33.2	61.0	27.0	,,	55.2	0.491	88.0	14.0
1505	7.2	32.8	135.9	33.0	69.0	27.4	3.5	59.0	0.462	87.8	14.0
1504	7.15	32.85	136.1	33.1	60.0	27.8	,,	59.1	0.461	87.8	14.1
1501	7.9	32.1	133.4	32.3	59.0	28.0	3.9	62·2	0.448	87.7	14.3
1503	8.65	31.35	130.7	31.7	58.0	27.3	4.4	64.0	0.443	87.3	14.2
1504	9.3	30.7	128.5	31.2	57.0	27.8	4.8	65.7	0.439	88.0	14.5
1507	10.5	29.5	124.3	30.2	63.0	27.9	5.0	68.3	0.436	87.2	14.8
1502	11.2	28.8	121.8	29.5	63.0	28.0	5.4	70.8	0.431	87.5	15.0
1500	12.7	27.3	116.4	28.2	61.0	28.4	5.5	73.8	0.433	88.0	15.0
1501	12.5	27.5	117.1	28.4	61.0	28.8	5.8	73.7	0.431	88.3	15.0
1500*	16.0	24.0	104.8	25.35	61.0	28.0	3.2	73.7	0.482		
1486	6.3	33.7	139.0	33.4	60.0	27.0	3.2	53.4	0.505	87.8	15.0
1513	6.3	33.7	139.0	33.9	60.0	26.8	3.2	49.5	0.537	88.0	15.1
1500	5.6	5.6	19.8	—	60.0	—	—	—	—	Motoring test	

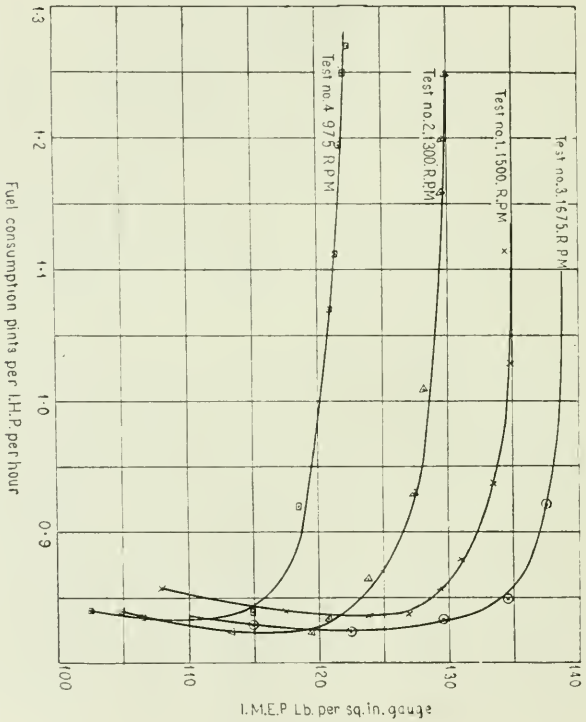
* Slightly unstable.

REMARKS.—Heating to air intake : 88 volts = 65 B.Th.Us. per minute (energy output varies as voltage squared.)

REMARKS ON TEST RESULTS.

With Low and High Compression : Series I.

It was observed that at all speeds, and both with high and low compression, the thermal efficiency obtained with alcohol was



SERIES I.

FIG. 38.—Relation between I.M.E.P. and consumption at different speeds.

Compression-ratio 3.8 : 1 ; heating to air intake 0.0435 B.Th.Us. per min. per r.p.m.

Ethyl alcohol 95 Vol. per cent.

Tests 1–4.

higher than that hitherto obtained, either with petrol at the lower compression-ratio or with benzole at the higher. In view of this fact, we thought it not only desirable to carry out a special series

of check tests on petrol at the lower compression-ratio, but also to measure and record the distribution of heat in all cases. We therefore arranged to measure the heat taken up by the cooling water during every test throughout Series I. and II. This necessarily involved a great deal of extra time and trouble, but the results obtained are very interesting.

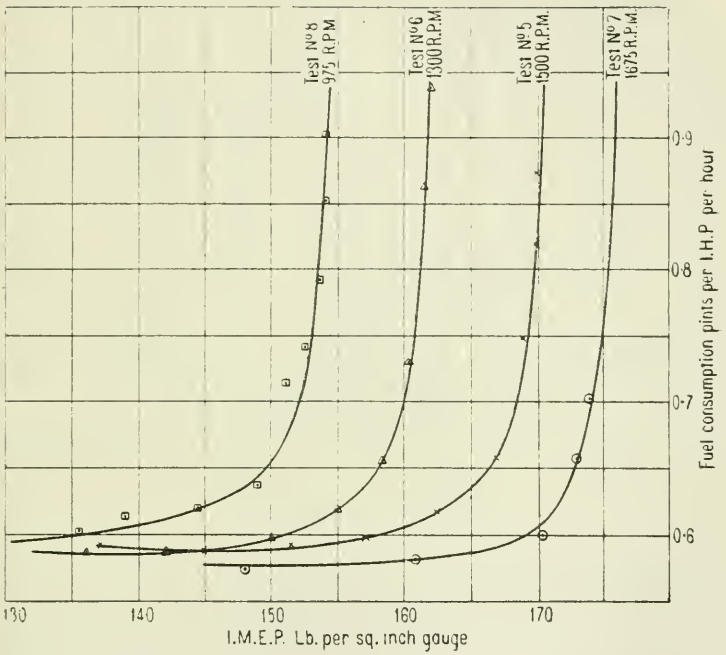


FIG. 39.—Relation between I.M.E.P. and consumption at different speeds.

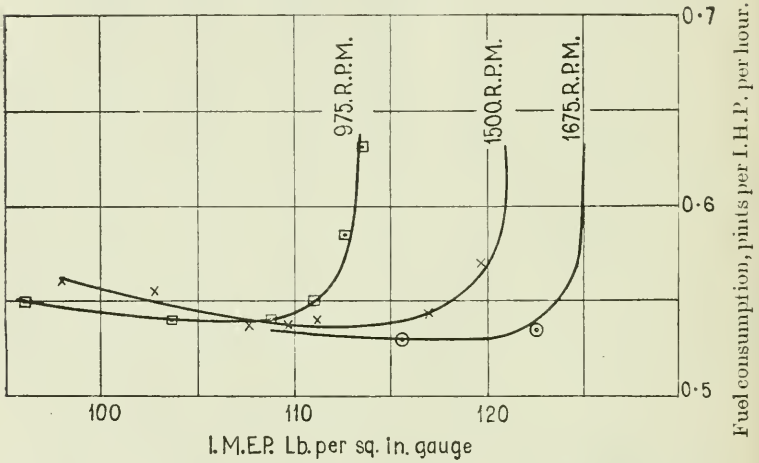
Compression-ratio 7 : 1 : heating to air intake 0.0435 B.Th. Us. per minute per r.p.m.
Ethyl alcohol 95 Vol. per cent.
Tests 5–8.

This series also included, as agreed, a check test on petrol "A" over the whole available range of mixture strength, and at a compression-ratio of 5.0 : 1. This test was run at the same jacket temperature and with the same heat input to the carburettor as in the case of the alcohol series. For convenience, however, it was run at a piston speed of 2000ft. per minute, or 1500 revs. per minute, instead of 1600ft. per minute.

The following observations may be noted :—

(1) That at all speeds the efficiency obtained with alcohol and with a compression-ratio of 3.8 : 1 is substantially greater than that obtained with petrol under precisely similar conditions ; the same applies at the high compression-ratio as between alcohol and benzole.

(2) That in both cases the efficiency is nearly independent of speed (rising slightly with increase of speed), and in both cases



Series I.

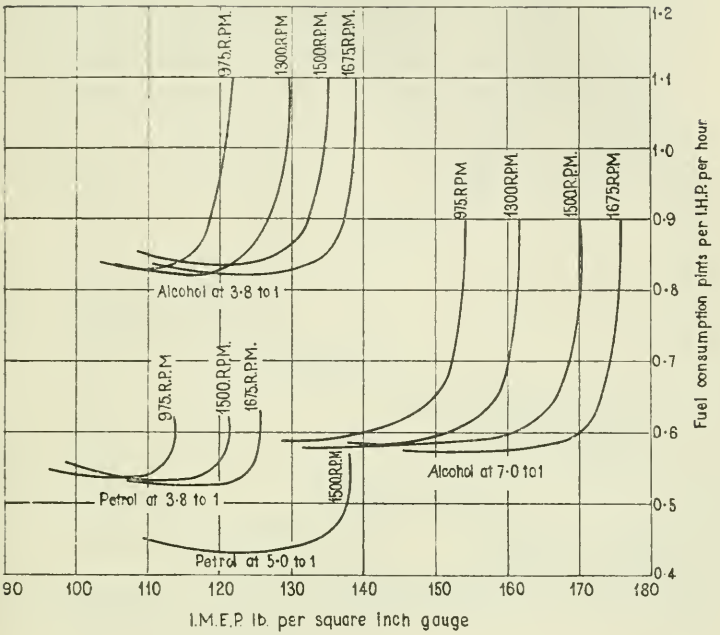
Fig. 40.—Relation between I.M.E.P. and fuel consumption at different speeds.

Compression-ratio 3.8 : 1 ; heating to air intake 0.0435 B.Th.Us. per min. per r.p.m.
Petrol "A."
Check test.

the power or mean pressure increases in approximately the same proportion with increase of speed.

(3) That the heat delivered to the jacket water is very substantially less in proportion to the total heat of the fuel with alcohol than with petrol, due to the lower temperature of the cycle, which accounts for the higher thermal efficiency observed. (See Heat-Balance Sheets and Fig. 42.)

(4) That relatively with petrol or benzole the thermal efficiency obtained with alcohol is equally high at high or low compression. (See Fig. 41.)



SERIES I.

FIG. 41.—Figs. 38–40 combined to show relative I.M.E.P. and consumption at different compression-ratios and speeds.

Ethyl alcohol 95 Vol. per cent and petrol "A."
Tests 1–8 and checks.

Minimum Fuel Consumption (Pints per I.H.P. per Hour).

R.p.m.	Petrol. R=3.8	Alcohol, R=3.8.	Alcohol. R=7.0.
975	0.538	0.835	0.593
1300	—	0.825	0.586
1500	0.536	0.835	0.590
1675	0.530	0.825	0.578

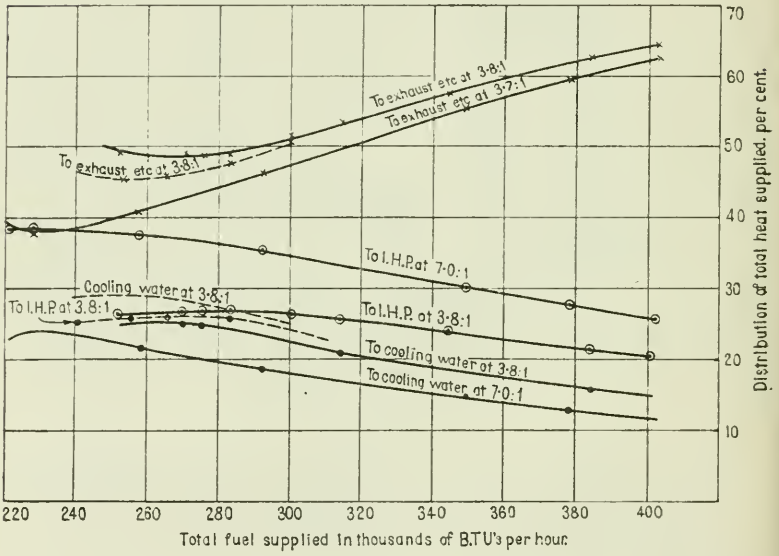
I.M.E.P. at 20 per cent Richer than at Minimum Fuel Consumption (lb. per sq. in.).

R.p.m.	Petrol. R=3.8	Alcohol. R=3.8.	Alcohol. R=7.0.
975	113.0	120.5	152.0
1300	—	128.5	160.0
1500	120.5	134.5	168.5
1675	125.0	138.5	174.0

Heating and cooling conditions practically the same for all curves.

(5) There is absolutely no indication that alcohol *requires* a high compression or is unsuitable for high speed with low compression. The curves both of power output and efficiency at any compression or at any speed run virtually parallel with those obtained from petrol, benzole, or any other hydro-carbon fuel, provided that the ignition timing is correctly adjusted for each fuel. (See Fig. 43.)

(6) When running with a compression-ratio of 7 : 1 and at speeds of over 1700 revs. per minute, occasional pre-ignition



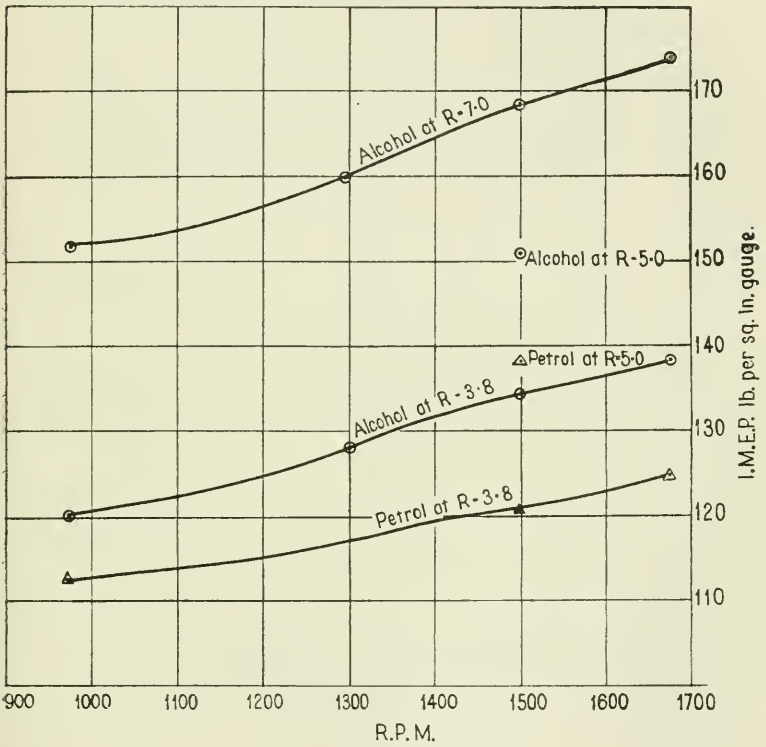
Series I.

Fig. 42.—Distribution of heat between I.H.P., cooling water, exhaust, etc., with varying mixture strength.

Full lines, ethyl alcohol 95 Vol. per cent.
Dotted lines, petrol "A."

occurred, but there were no signs of detonation. In view of the great increase in the maximum pressure (and the constant destructive effect of pre-ignition) and the relatively small gain in economy with further increase in compression, it is very doubtful whether it will ever be desirable to operate with a compression-ratio in excess of 7 : 1.

The test sheets of Tests Nos. 1 to 8 and the Heat-Balance Sheets of Tests Nos. 1 to 6 and No. 8, give all the readings taken during this series. The curves in Figs. 38, 39, 40 and 41 show the results given in the test sheets plotted in terms of indicated mean pressure



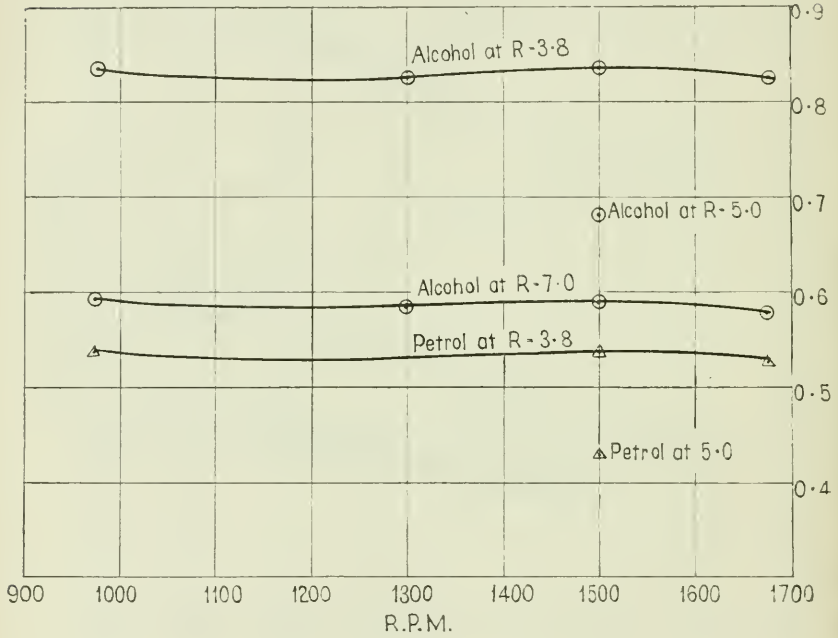
Series I.

Fig. 43.—Variation of I.M.E.P. with speed at different compression-ratios at full throttle. Temperature and mixture strength similar.

Heating to air intake 0.0435 B.Th.Us. per min. per r.p.m
Ethyl-alcohol 95 Vol. per cent and petrol " A."
Tests 1–8 and check.

and fuel consumption in pints per indicated horse-power per hour reduced to 15° C. The curve in Fig. 42 gives the heat distribution with varying mixture strength, and is a summary of the heat-balance sheets. Fig. 44 shows the variation in fuel consumption over a range

of speed of from 975 to 1675 revs. per minute at compression-ratios of 3.8 : 1 and 7 : 1. Fig. 43 shows the variation in mean pressure over a range of speed of from 975 to 1675 revs. per minute, the mean pressure in each case being that obtained with a mixture strength 15 per cent richer than the most economical setting. Fig. 45 shows



Series I.

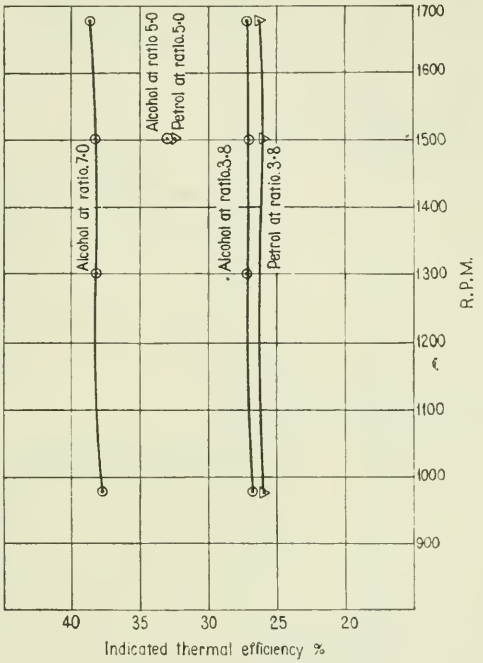
Fig. 44.—Variation of minimum consumption with speed at different compression-ratios. Full throttle.

the variation of indicated thermal efficiency with speed at different compression-ratios.

Throttle Tests Varying Load : Series II.

This series requires no comment, for, as will be seen from the test sheets and from Figs. 48, 48A, 48B, and 48C, the behaviour of petrol and alcohol under throttled conditions was identical in every respect.

In this series, as in Series I., we deemed it advisable to take readings of the complete heat-balance in every instance. As explained previously, we did not run the check test on petrol at the lower speed, but from data which we already have, we are satisfied that the relation between the behaviour of alcohol and that of petrol



Series I.

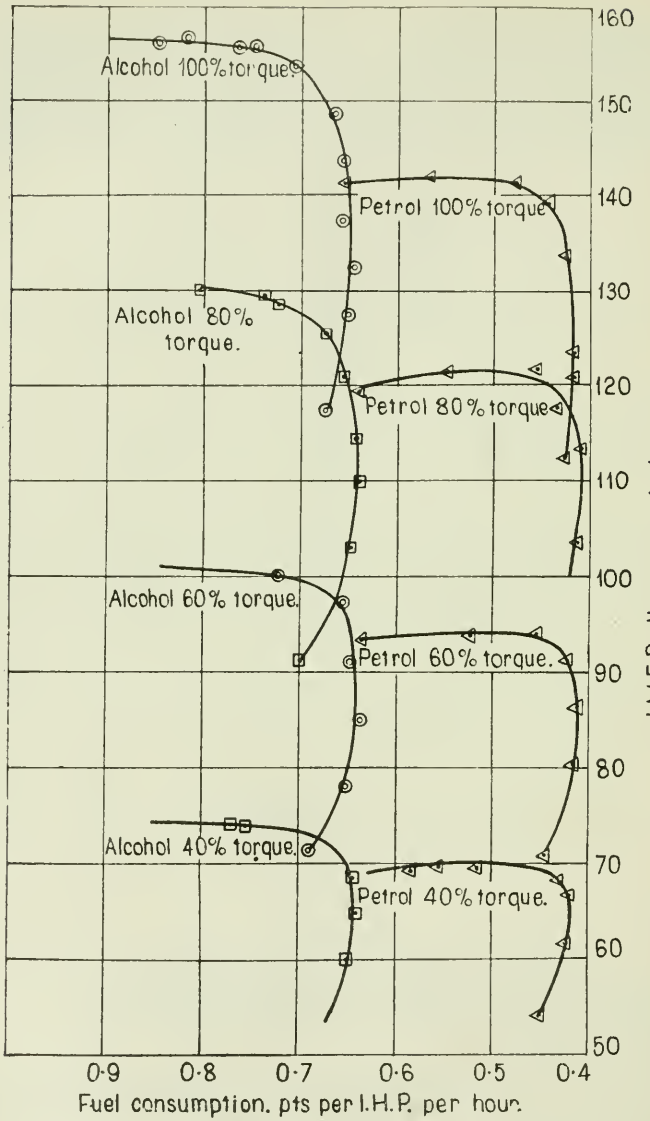
Fig. 45.—Variation of indicated thermal efficiency with speed at different compression-ratios, full throttle.

Gross calorific value of alcohol = 11,130 B.Th.Us. per lb.
Gross calorific value of petrol "A" = 18,580 B.Th.Us. per lb.
Tests 1–8 and checks.

is the same at the lower speed as at the higher speed, as indeed might be expected.

With Hot and Cold Jacket Water (*Figs.* 49 *and* 51) *: Series III.*

This series was run with a constant heat input to the carburettor of 65 B.Th.Us. per minute, 0.0433 B.Th.Us. per revolution per



Series II.

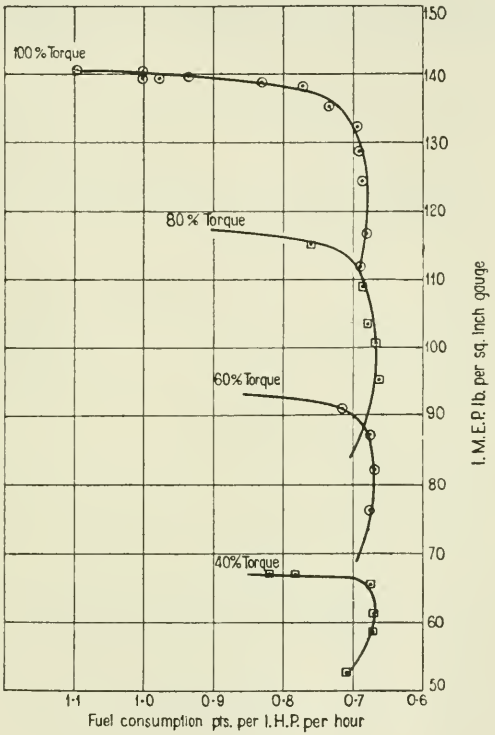
Fig. 46.—Variation of I.M.E.P. and consumption with variation in throttle opening.

Compression-ratio 5.45 : 1; revs. per minute 1500; heating to air intake proportional to throttle opening and equal to 65 B.Th.Us. per min. at full torque.

Alcohol 95 Vol. per cent and petrol "A."

Tests 1–4 and check tests. 1–4.

minute, and under identical conditions in every respect, except as to the temperature of the jacket water. Both sets of tests were carried out during one continuous run, throughout which the speed was



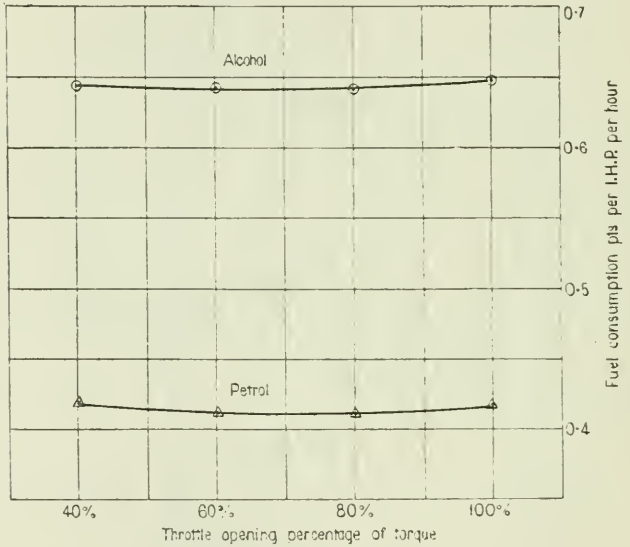
Series II.

Fig. 47.—Variation of I.M.E.P. and consumption with variation in throttle opening.

Compression-ratio 5.45 : 1 ; revs. per minute 950 approx.; heating to air intake proportional to throttle opening and equal to 40 B.Th.Us. per min. at full torque.
Alcohol 95 Vol. per cent.
Tests 5–8.

maintained constant, and the engine run with wide-open throttle. During the hot test, which was run first, the water circulation was adjusted to give the highest temperature that could be maintained

without boiling, while during the cold test the maximum flow of cooling water was employed. It will be observed that the indicated horse-power obtained during the cold test was about 6 per cent greater than during the hot test, simply on account of the greater weight of charge taken into the cylinder. The consumption per indicated horse-power per hour at most economical load was found to be 0.694 pints during the hot-jacket test, and 0.688 pints during



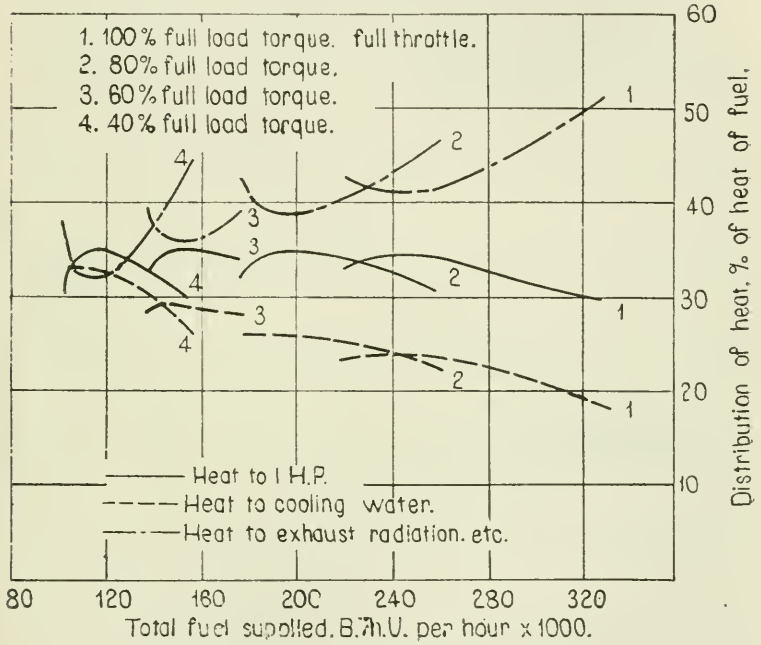
Series II.

Fig. 48.—Relation between minimum consumption and throttle opening.

Compression-ratio 5.45 : 1 ; heating proportional to maximum power under each condition.
Alcohol 95 Vol. per cent and petrol "A."
Tests 1–4 and checks.

the cold-jacket test. Owing to the very low piston friction in this engine, the actual brake horse-power was also considerably greater during the cold-jacket test, while the motoring tests revealed the fact that the total mechanical losses varied only by the equivalent 3 lb. mean effective pressure as between the two conditions. In many, if not in most car or commercial-vehicle engines, the piston friction is so severe and so dependent upon the temperature, that

it is very probable that the brake horse-power might be actually lower with cold than with hot jacket water, even though for this to be the case, the increase in piston friction as between cold and hot jacket water would have to be equivalent to no less than 8.5 lb. per sq. in. mean pressure, the difference in the maximum indicated mean pressure observed during these two tests. From these tests

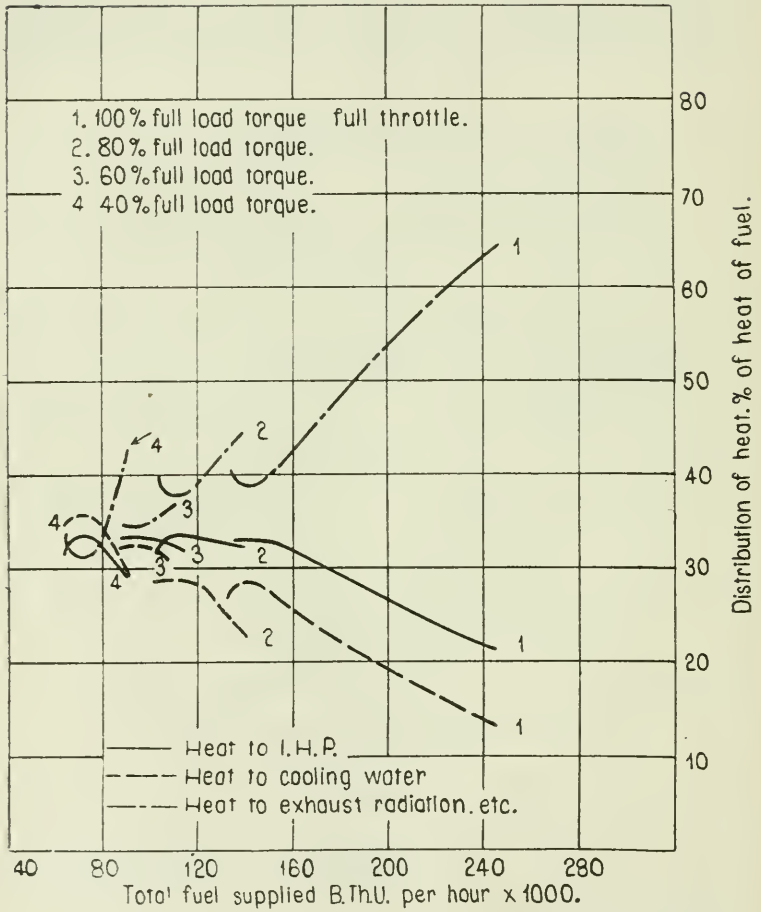


SERIES II.—Heat-balance.

FIG. 48 (A).—Distribution of total heat between I.H.P., water and exhaust.

Mixture strength varied as indicated by the horizontal ordinate.
Ethyl alcohol 95 Vol. per cent.
Tests 1–4.

it is quite clear that the result of raising the jacket water temperature is merely to reduce both the power output and efficiency, as is the case with petrol, benzole, or any other reasonably volatile fuel. In no instance has any gain, either in indicated power or efficiency, been observed with increase of jacket temperature in the case of fuels boiling below 200° C.



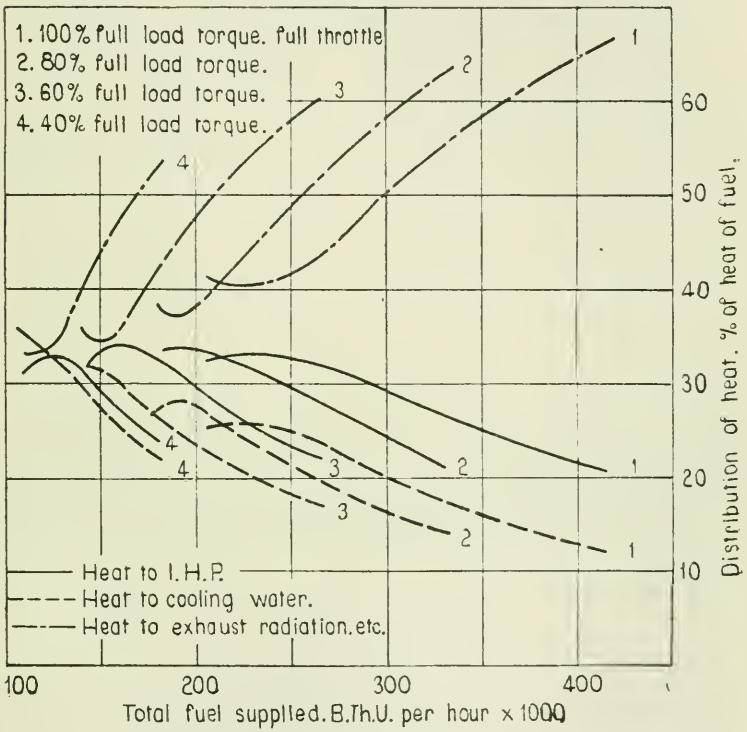
SERIES II.—Heat-balance.

FIG. 48 (B).—Distribution of total heat between I.H.P., water and exhaust.

Mixture strength varied as indicated by the horizontal ordinate.
Ethyl alcohol 95 Vol. per cent.
Tests 5–8.

With Heated and Unheated Carburettor (Figs. 50 and 51) : Series IV.

It will be observed that when the heat input to the carburettor is equivalent to 100 B.Th.Us. per minute, the power output with a mixture strength giving a consumption of 0.8 pints per indicated



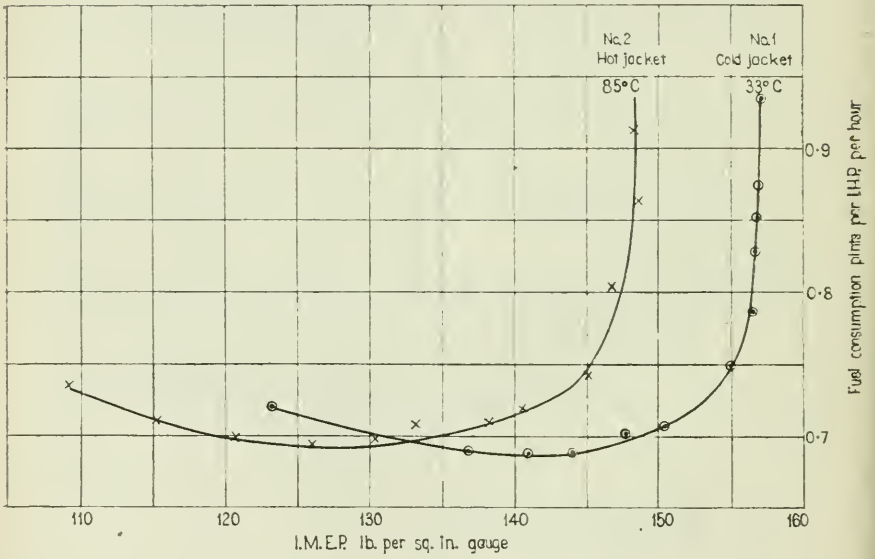
Series II.—Heat-balance.

Fig. 48 (c).—Distribution of total heat between I.H.P., water and exhaust.

Mixture strength varied as indicated by the horizontal ordinate.
Petrol " A."
Check Tests 1–4.

horse-power per hour drops 7.2 per cent, as compared with no heat input, but the efficiency rises about 1 per cent, due largely to a reduction in the condensation losses. In the case of a multiple-cylinder engine, the difference in efficiency as between these two

conditions of heating would be greater, though how much greater must depend largely on the design of the carburettor, and more particularly that of the induction system. The greater power output obtained with no heating is due, of course, to the greater density of the charge. Attention should be drawn to the measurement of the gas temperatures in the induction pipe in these two tests. Although the absolute values of these temperatures cannot be relied upon, because the thermometer behaves to some extent as a wet-bulb ther-



Series III.

Fig. 49.—Influence of temperature of cooling water on consumption and I.M.E.P. at full throttle opening.

Compression-ratio 5 : 1; revs. per minute 1500; heat to air intake 65 B.Th.Us. per minute.
Ethyl alcohol 95 Vol. per cent.
Tests 1 and 2.

mometer, yet the relative figures are a fair indication of the rate of evaporation of the alcohol. In spite of the fact that in the case of Series IV., heat was being applied at the rate of 100 B.Th.Us. per minute, the temperature of the gas was only 11 degrees higher than when no heat was applied. Compared with the tests on full throttle in Series II., corrected for a compression-ratio of 5 : 1, and also the

tests in Series III., with a heat input of 65 B.Th.Us. per minute, with approximately the same jacket temperature, and at the same speed and compression, the results are as in Table XIII. :—

TABLE XIII.

Series	Jacket water (° C.)	Heat input (B.Th.Us. per minute)	Minimum fuel consumption (pint per I.H.P. per hr.)	Indicated mean pressure with consumption (0.8 pint per I.H.P. per hr. (lb. per sq. in.)
II.	60	65	0.679	151.0
III.	85	65	0.694	147.3
	35	65	0.688	156.6
IV.	60	100	0.676	145.8
	60	Nil	0.696	157.1

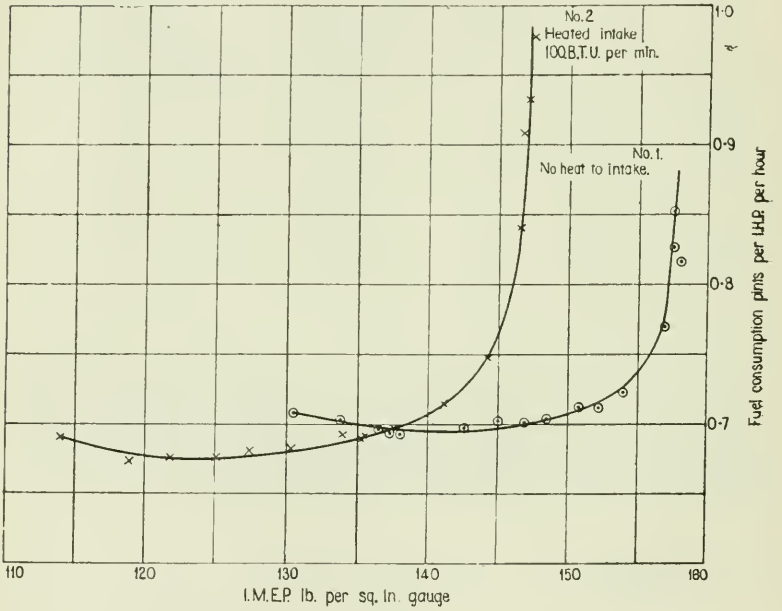
Series III. and IV. indicated that the thermal efficiency rises slightly with increase of heat input to the carburettor, but falls with increase of heat to the jacket water. In either case, the addition of heat to the working fluid from either source reduces the power output obtainable, as, indeed, would be expected.

Notes on the General Behaviour of the Engine when Running on Ethyl Alcohol.

At all times and under all conditions the running of the engine was noticeably sweeter and smoother than on petrol, and this applies even to the highest compression used, namely, 7 : 1, corresponding to a compression-pressure of approximately 185 lb. per sq. in. No trace of detonation occurred under any circumstances, but during the tests in Series I., with a compression-ratio of 7 : 1, pre-ignition set in after a prolonged run at 1675 revs. per minute. The pre-ignition when it occurred was of an extremely violent nature, as, indeed, might be expected at so high a compression-ratio. Examination of the plug points showed nothing to indicate the cause. In this respect, namely, its tendency to pre-ignite without any preliminary warning in the way of detonation, the behaviour of alcohol resembles that of the various members of the aromatic series, and certain other fuels we have tested. It was noted that under all conditions alcohol requires a few degrees more ignition advance than does petrol, and this becomes the more noticeable when working with weak mixtures.

Throughout the tests there has been no indication of any harmful effects, nor has the power output or efficiency of the engine varied measurably between the start and finish.

It may be of interest to note that on several occasions the engine



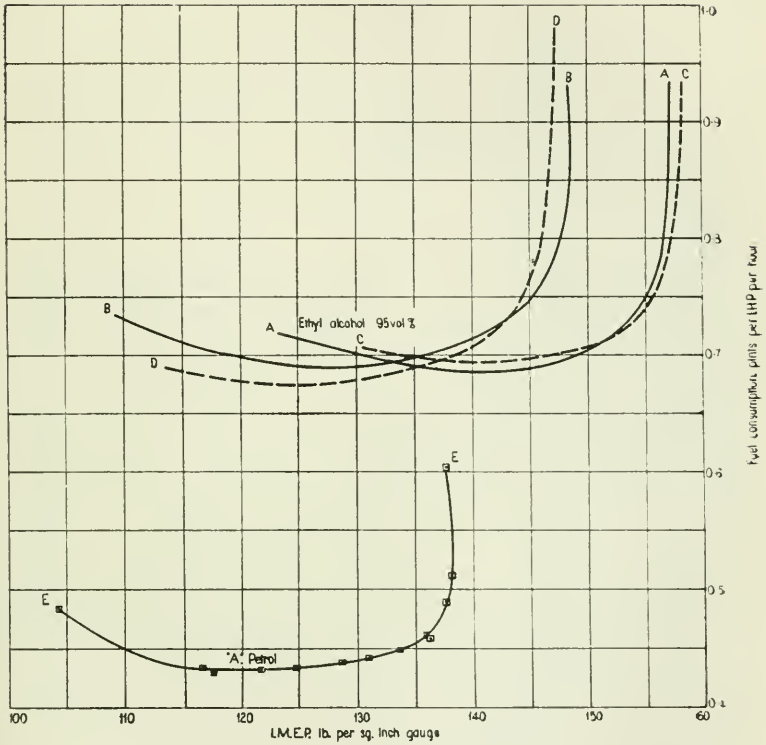
Series IV.

Fig. 50.—Influence of heat to intake on consumption and I.M.E.P. at full throttle opening.

Compression-ratio 5 : 1 ; revs. per minute 1500 ; jacket outlet temperature 60° C.
Ethyl alcohol 95 Vol. per cent.
Tests 1 and 2.

started up from " all cold " on ethyl alcohol, and that it invariably started at once after standing several hours between tests.

A practical difficulty may arise during any transition period, when changing over from alcohol to petrol, due to the deposition of water in the system.



SERIES III. AND IV.

FIG. 51.—Influence of heat to air intake and of circulating water on consumption and I.M.E.P. at full throttle opening.

Compression-ratio 5 : 1 ; revs. per minute 1500.

Curve.	Series.	Heat to intake, B.Th.Us. per min.	Jacket temp., deg. C.
A	III., No. 1	65	33
B	III., No. 2	65	85
C	IV., No. 1	0	60
D	IV., No. 2	100	60
E	III., Check	65	60

TESTS ON 90 VOL. PER CENT AND 99 PER CENT ALCOHOL.

General Notes.

Since the tests reported in the preceding pages, the experimental engine has had a serious breakdown. When running some experiments with a very high compression-ratio, the connecting-rod split suddenly, wrecking the piston, cylinder, and the whole of the valve gear. It has therefore been necessary to dismantle the engine entirely and to fit a new cylinder, connecting-rod, piston, and valve mechanism. Although every possible care has been taken to restore exactly the conditions which obtained before the accident, the fitting of new valves, cylinder, piston, and piston rings has necessarily altered slightly the calibration of the engine, and although before commencing this second batch of tests the engine had been run for between 50 and 60 hours, neither the power output nor the efficiency on standard fuels is quite so high as it was previously. The difference is, however, exceedingly small, for the slightly reduced power output obtained is due largely to the very high atmospheric temperature which prevailed at the time these experiments were run. When due allowance is made for the higher atmospheric temperature conditions prevailing during these tests as compared with those reported previously, it will be found that the performance of the engine has depreciated to the extent of between 0.5 per cent and 1 per cent. This fact unfortunately prevents direct comparisons being made between the test results now submitted and those reported previously; we had, however, just sufficient of the 95 Vol. per cent alcohol left to enable us to carry out check tests, and these are shown in the test sheets and curves of the first and second series, namely, at compression-ratios of 3.8 : 1 and 6 : 1.

The apparatus employed and the procedure adopted throughout was precisely the same as in the previous tests, with the exception that in the case of one set of tests on each fuel in Series III. we brought into operation our new air-measuring device, referred to on p. 134 and illustrated in Figs. 52 and 53.

The tests in Series I. were commenced with a compression-ratio of 7 : 1, but after running for some time some violent pre-ignitions occurred, due to scale on the sparking-plug points. In view of the

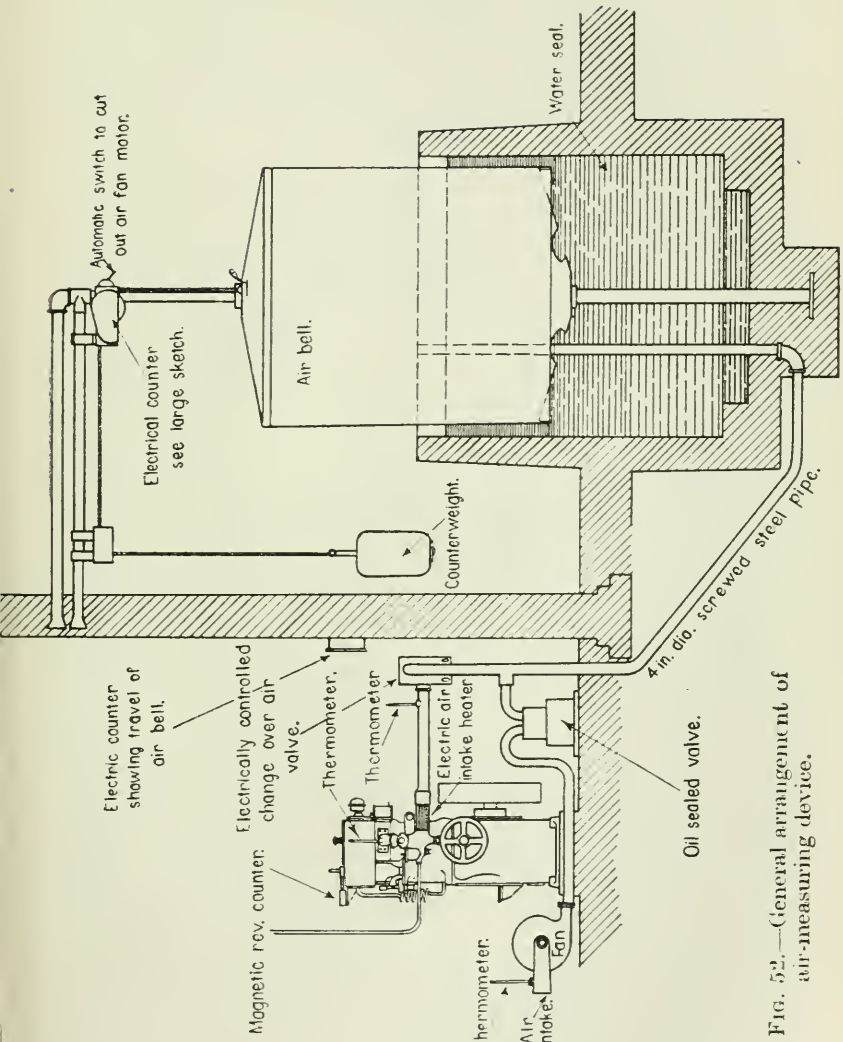


FIG. 52.—General arrangement of air-measuring device.

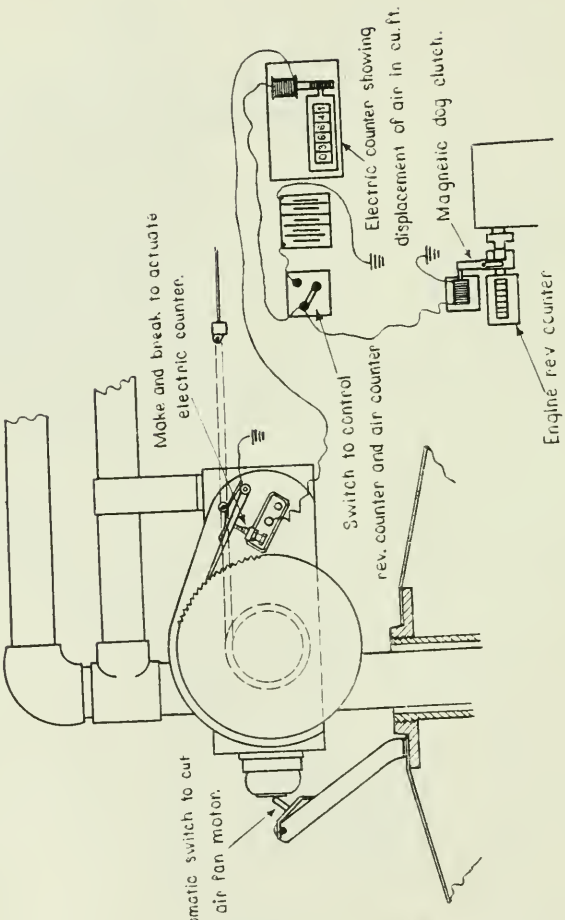

Make and break to actuate electric counter.
Electric counter showing displacement of air in cu.ft.
Magnetic dog clutch.
Switch to control rev. counter and air counter
Engine rev counter
Automatic switch to cut cut air fan motor.

FIG. 53.—Arrangement of electric counters for air-measuring device.

accident to the engine referred to above and owing to the very violent nature of the pre-ignitions (which are always liable to occur at so high a compression-pressure), we deemed it advisable to reduce the compression-ratio to 6 : 1 for this Series.

In our first report, p. 112, we explained the method we adopted in arriving at the effective calorific value of the 95 Vol. per cent alcohol, a method to which we have adhered in the case of the 90 Vol. per cent and the 99 per cent samples. Following this reasoning, we arrive at the values shown in Table XIV. :—

TABLE XIV.

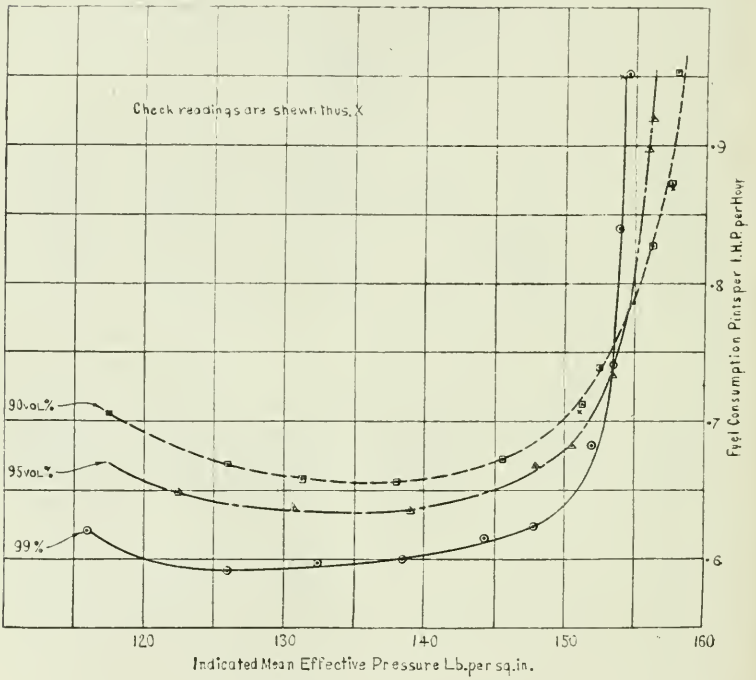
Nominal strength of fuel. Per cent.	Specific gravity at 15° C.	Ethyl alcohol by weight. Per cent.	Water content by weight. Per cent.	Latent heat of alcohol. B.Th.Us. per lb.	Total latent heat. B.Th.Us. per lb.
90 Vol.	0.8315	86.5	13.5	343	474
95 Vol.	0.8155	92.5	7.5	367	440
99	0.7955	99.4	0.6	394	400

Nominal strength of fuel. Per cent.	Calorific value (ex latent heat). B.Th.Us. per lb.	Add 94 per cent latent heat of alcohol. B.Th.Us. per lb.	Effective calorific value. B.Th.Us. per lb.	Effective calorific value. B.Th.Us. per pint.	Gross latent heat of evaporation B.Th.Us. per pint.
90 Vol.	10,080	320	10,400	10,810	492
95 Vol.	10,780	345	11,125	11,340	449
99	11,580	370	11,950	11,890	398

Description of Apparatus.

For the measurements of air consumption we made use for the purpose of these tests of a special air-measuring device (Figs. 52 and 53), consisting of a small balanced and water-sealed gas holder having a total capacity of approximately 120 cu. ft. and containing, therefore, about 100 per cent excess air capacity for the combustion of $\frac{1}{4}$ pint of fuel at the weakest mixture strength. The air intake to the carburettor is controlled through the medium of an electrically-operated double-beat valve, by means of which the air supply to the engine can be changed over instantaneously, either from the outside air or from the gas holder. When drawing air from the gas holder the consumption is recorded automatically in the following manner. From the top of the holder a chain passes over a small sprocket wheel, on the spindle of which is also mounted

a large-diameter ratchet wheel ; a pawl bears against this latter, and as the gas holder falls and the sprocket and ratchet wheels rotate, the pawl makes and breaks contact on a low-voltage supply (Fig. 53). The number of teeth on the sprocket and ratchet wheels have been so chosen that each complete make and break of contact by the



Series I.

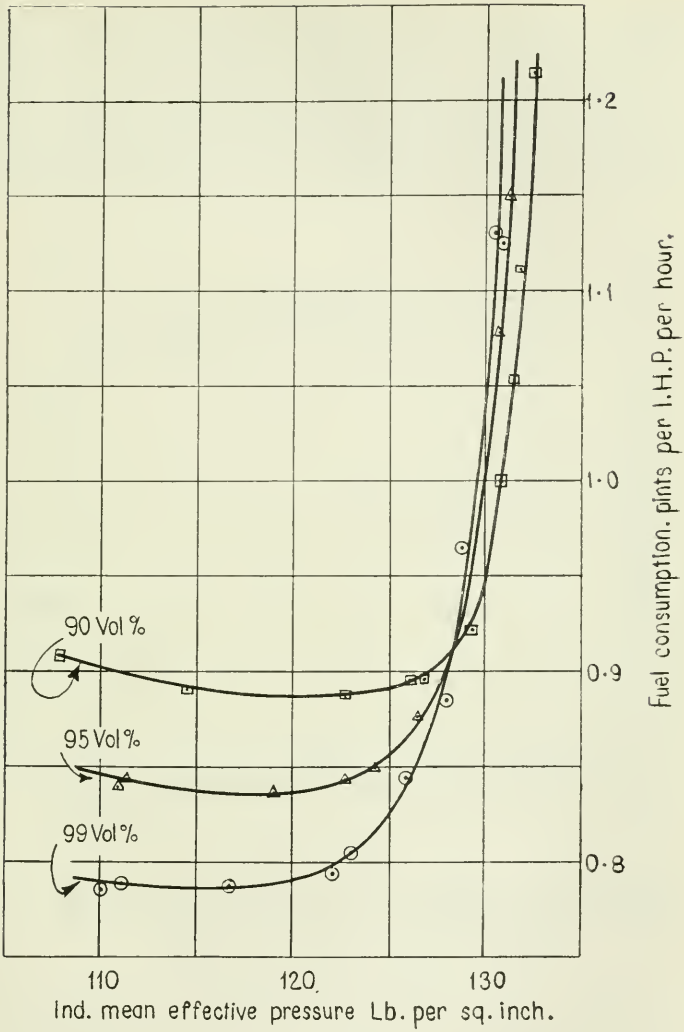
Fig. 54.—Variation of I.M.E.P. and consumption with mixture strength.

Compression-ratio 6 : 1 ; revs. per minute 1500 : jacket temperature 60° C.; heat to air intake 65 B.Th.Us. per minute (0.0433 B.Th.Us. per revolution).

Alcohol 99 Vol. per cent, 95 Vol. per cent, and 90 Vol. per cent.

Tests 1 and 2 and check.

pawl corresponds with a displacement of exactly one-fifth of a cu. ft. in the holder. On the operator's desk is mounted an ordinary revolution counter, also operated by means of a pawl and ratchet ; in this case, however, the pawl is operated by means of an electro-

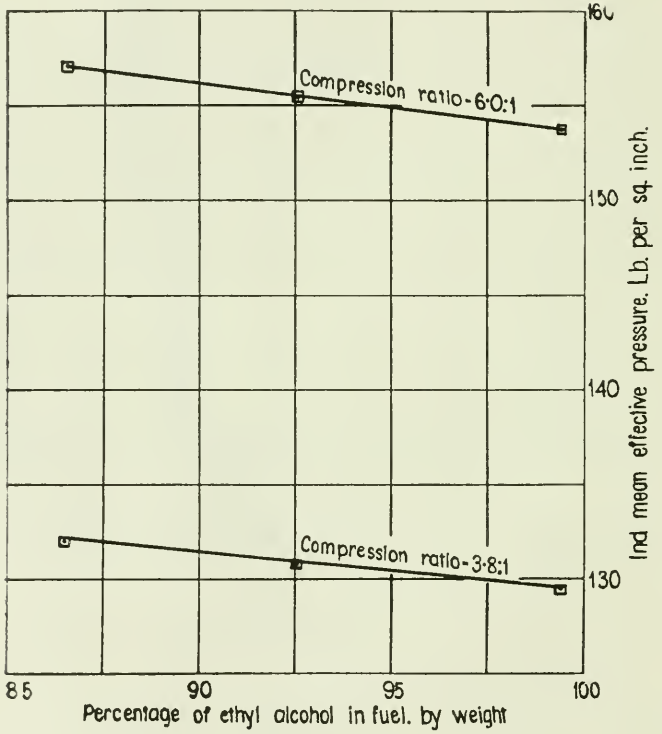


SERIES II.

FIG. 55.—Variation of I.M.E.P. and consumption with mixture strength.

Compression-ratio 3.8 : 1 ; 1500 revs. per minute ; jacket temperature, 60° C.; heating to air intake 65 B.Th.Us. per minute.
Alcohol 90, 95, and 99 Vol. per cent.
Tests 1 and 2 and check.

magnet in circuit with the contact-maker on the holder. The counter therefore records exactly the consumption of air in one-fifths of a cu. ft. The air-recording counter and the magnetically-controlled revolution counter on the engine are both operated by the same low-pressure circuit and controlled by the same switch, so that both the revolutions of the engine and its air consumption during that number of revolutions are simultaneously recorded.

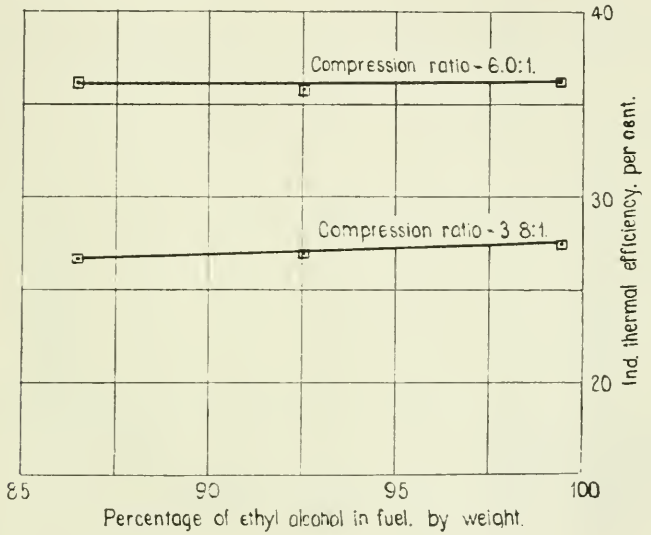


Series I. and II.

Fig. 56.—Variation of I.M.E.P. with different concentrations of ethyl alcohol and water for high and low compression-ratios.

The air holder is filled by means of an electrically-driven rotary blower which delivers air through a non-return valve in the form of an oil seal. This blower is started after the completion of a consumption test and is stopped automatically so soon as the air holder is filled. Thermometer pockets are provided on the air intake to

the fan and also on the air passage from the holder to the engine, and the temperature of the air is recorded. Care is taken to fill the holder only a few seconds before each test is taken, and since the normal duration of a test at full power is only about 50 seconds it follows that the holder is filled and exhausted in considerably less than 2 minutes. Under ordinary weather conditions the temperature-difference of the air before and after entering the holder does not exceed 1 degree C., and even under the most extreme conditions



SERIES I. AND II.

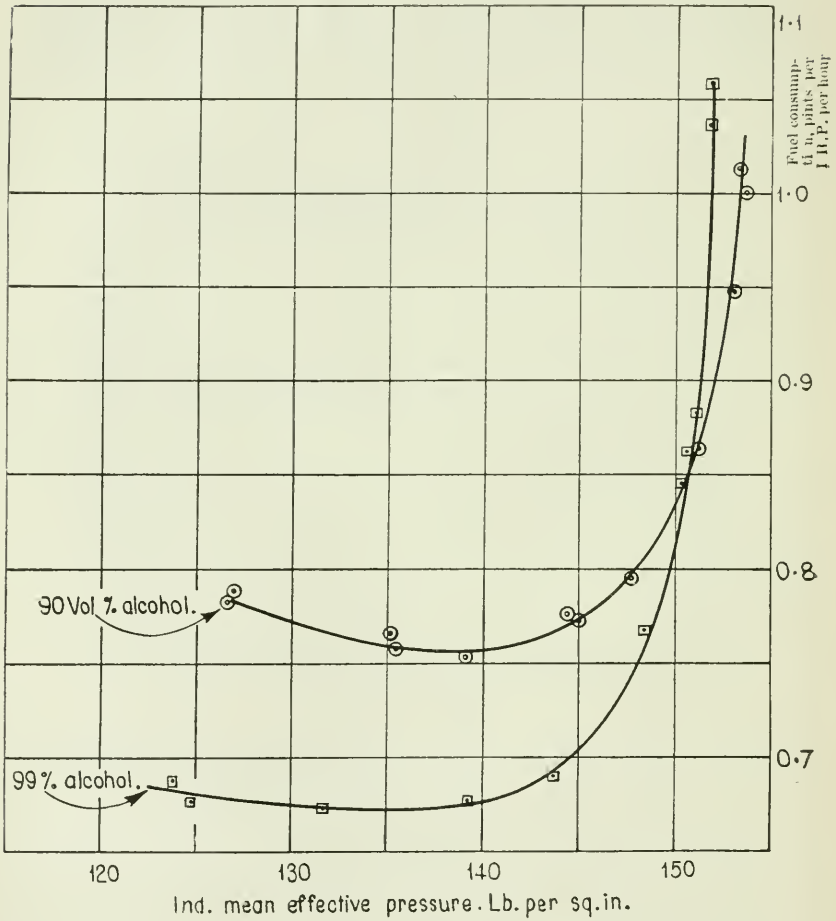
FIG. 57.—Variation in thermal efficiency with different concentrations of ethyl alcohol and water for high and low compression-ratios.

and with the most intense sunlight on the holder it does not exceed 2 degrees C. when the tests are carried out reasonably quickly.

Series I. and II.

For reasons already explained this series was run at compression-ratios of 6 : 1 and 3.8 : 1 instead of 7 : 1, as had orginally been employed in the previous series of tests already reported upon. In view of the fact that the calibration of the engine generally had been somewhat disturbed by its breakdown, and also because no previous tests had been run at a compression-ratio of 6 : 1 on

95 Vol. per cent ethyl alcohol, we deemed it desirable, for comparative purposes, to run afresh a directly comparable series on 95 Vol. per cent ethyl alcohol, of which just sufficient remained from

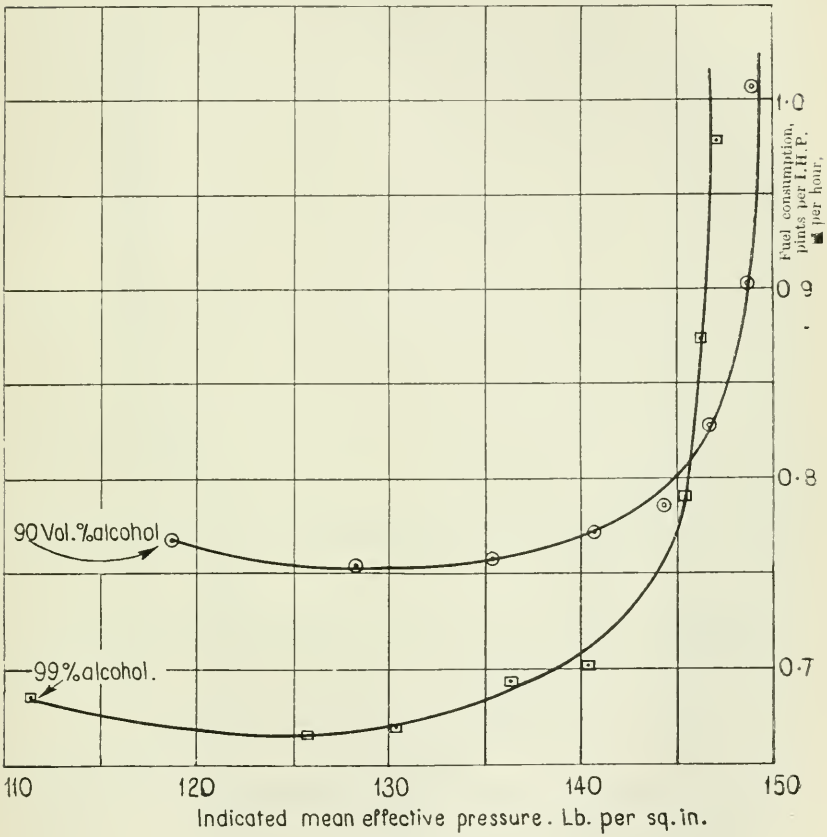


SERIES III.

FIG. 58.—Variation of I.M.E.P. and fuel consumption with mixture strength.

Compression-ratio 5 : 1 ; no heating to air intake.
Alcohol, 90 and 99 Vol. per cent.
Test 1.

the last barrel supplied to us. From an academic, if not also from a practical point of view, this series has proved particularly interesting. Apart from the obvious one of their different calorific values, the



SERIES III.

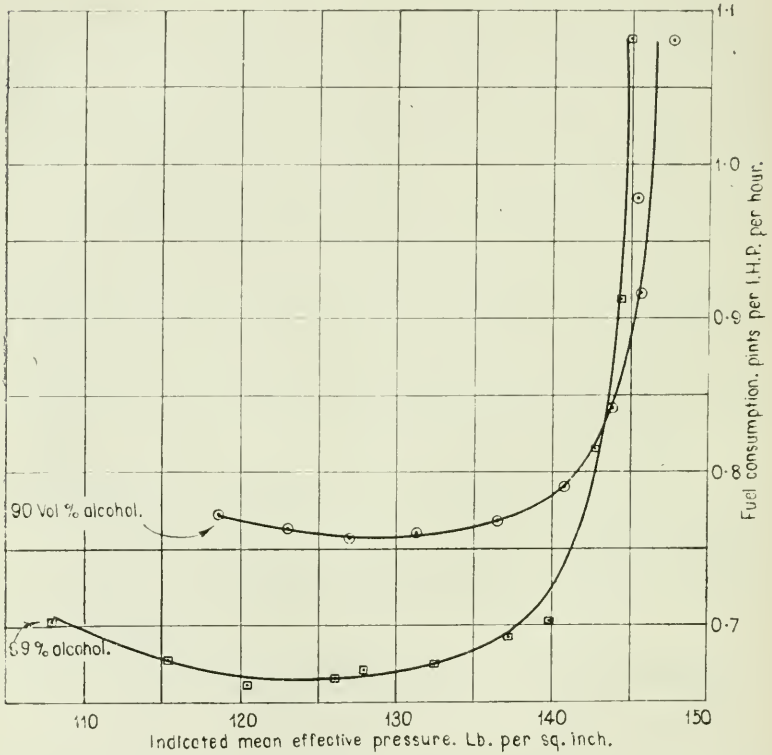
FIG. 59.—Variation of I.M.E.P. and consumption with mixture strength.

Compression-ratio 5 : 1; heating to air intake 40 B.Th.Us. per minute. Alcohol 90 and 99 Vol. per cent.
Test 2.

outstanding difference between these fuels lies in their different latent heats of evaporation, which amount to approximately 400, 440 and 474 B.Th.Us. per lb. respectively for the 99 per cent,

95 Vol. per cent and 90 Vol. per cent strengths, or 398, 449 and 492 B.Th.Us. per pint respectively.

As pointed out in our previous report, the whole of the heat



SERIES III.

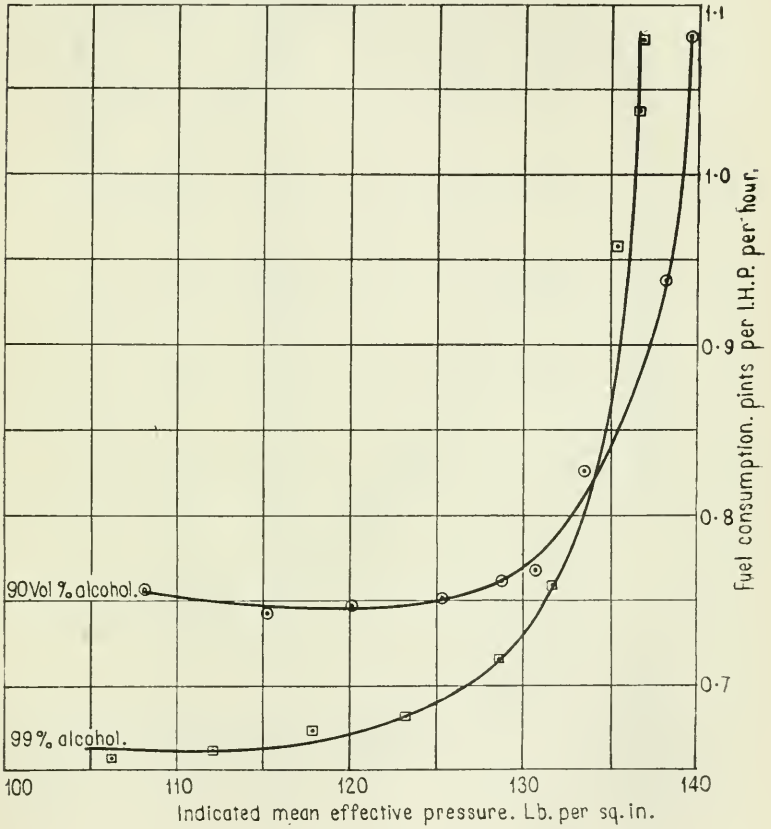
FIG. 60.—Variation of I.M.E.P. and consumption with mixture strength.

Compression-ratio 5 : 1 ; heating to air intake 60 B.Th.Us. per min.
Alcohol 90 and 99 Vol. per cent.
Test 3.

required to overcome the latent heat of evaporation of the fuel is supplied :

(*a*) By the heat supply to the air intake.
(*b*) By contact with the hot walls of the combustion chamber, piston, valves, etc.
(*c*) By admixture with the hot residual exhaust products.
(*d*) By the heat of compression.

With the exception of (*a*) and (*d*) the direct effect of a high latent heat of evaporation is to increase the weight of working fluid retained in the cylinder by lowering the temperature of the gases therein during

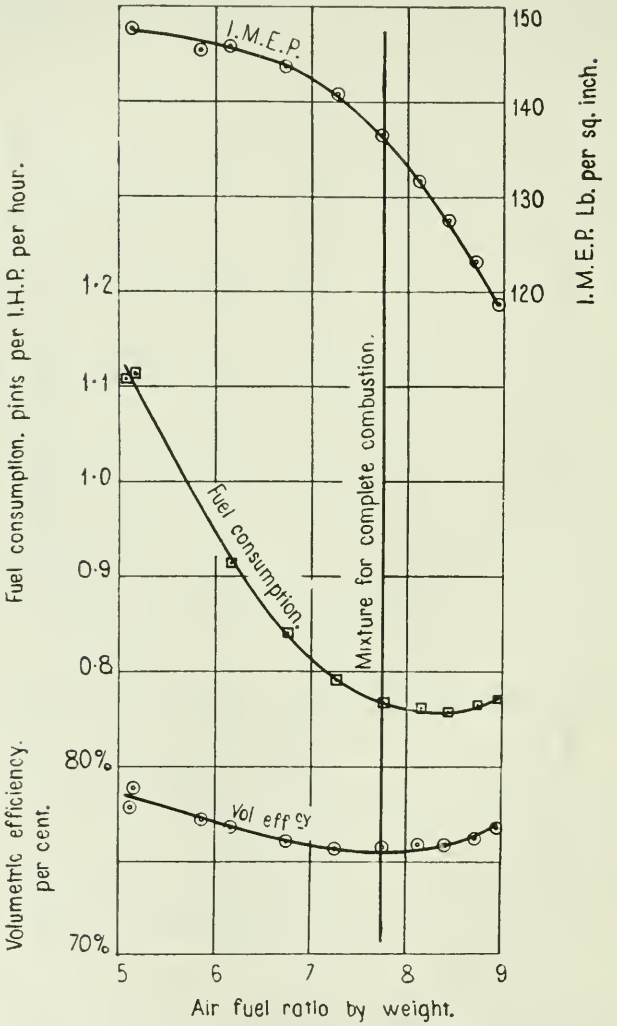


Series III.

Fig. 61.—Variation of I.M.E.P. and consumption with mixture strength.

Compression-ratio 5 : 1 ; heating to air intake 120 B.Th.Us. per min. Alcohol 90 and 99 Vol. per cent.
Test 4.

the period of the suction stroke. So long, therefore, as the thermal efficiency remains the same, *i.e.*, so long as the same proportion of the heat of combustion of the fuel is converted into useful work, the

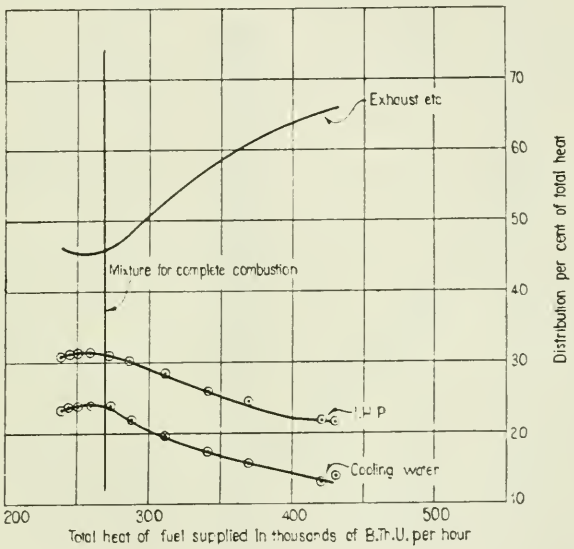


Series III.

Fig. 62.—Variation of volumetric efficiency, I.M.E.P. and consumption with mixture strength.

Compression-ratio 5 : 1 ; heating to air intake 60 B.Th.Us. per minute.
Alcohol 90 Vol. per cent.
Test 3.

power output will be proportional to the absolute temperature of the gases retained in the cylinder at the end of the suction stroke, and this in turn will be governed very largely by the latent heat of evaporation. Provided, therefore, that the thermal efficiency is the same, we might reasonably expect to find an increase in mean pressure as the water content is increased, so long as the whole of the fuel and water is completely evaporated before the end of the com-

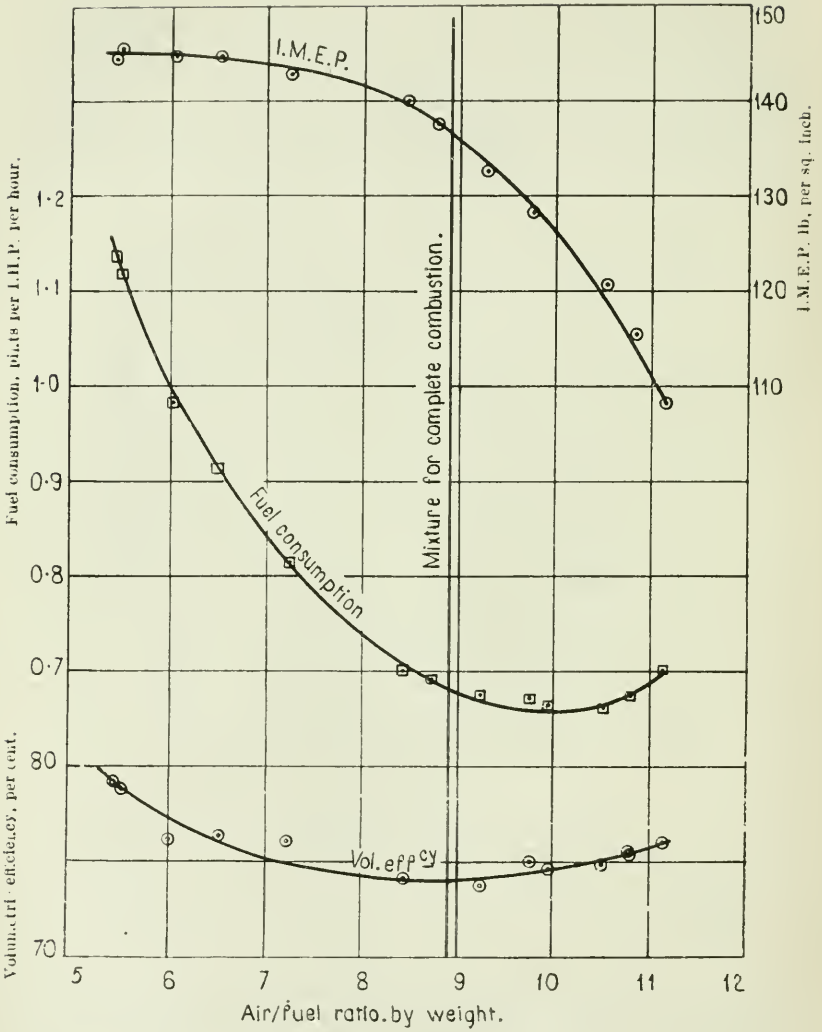


Series III.

Fig. 63.—Variation in distribution of heat with mixture strength.

Alcohol 90 Vol. per cent.
Compression-ratio 5 : 1 ; heating to air intake 60 B.Th.Us. per minute.
Test 3.

pression stroke. If, however, any water remains in a liquid state at the time of ignition, then it is clear that some part of the heat of combustion must be devoted to overcoming the latent heat of evaporation and a direct loss both in power and efficiency must result therefrom. We may therefore expect to find that, as the water content, and therefore the latent heat of the fuel, is increased, there will be a nearly corresponding increase in power output until a certain critical point is reached when any further addition of water will result in a rapid diminution in power and efficiency. Examined in this light, the results show that even in the case of the 90 Vol.



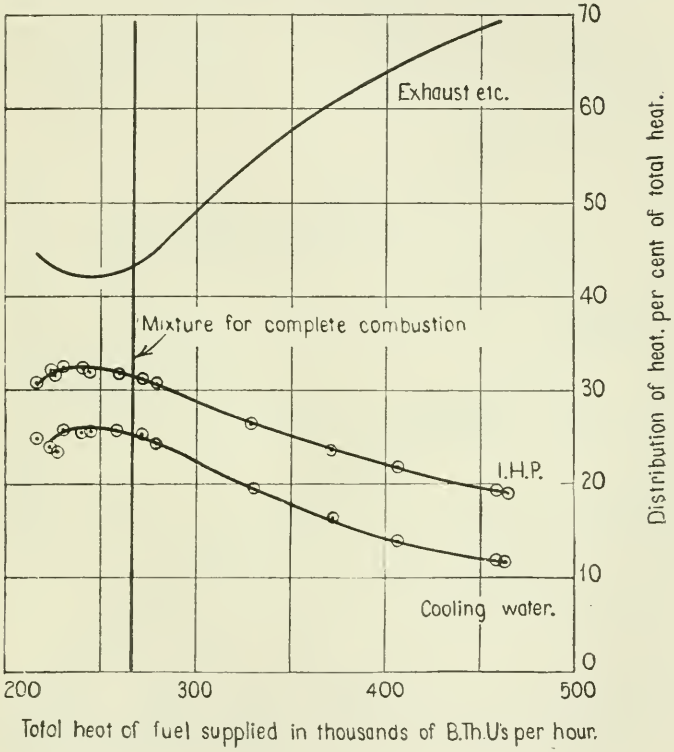
SERIES III.

FIG. 64.—Variation of volumetric efficiency, I.M.E.P. and consumption with mixture strength.

Compression-ratio 5 : 1 : heating to air intake 60 B.Th.Us. per minute
Alcohol 99 Vol. per cent.
Test 3.

per cent alcohol this critical point was never reached even with mixture strengths more than 30 per cent rich.

Although at a mixture strength about 20 per cent in excess of that giving complete combustion the percentage increase in mean



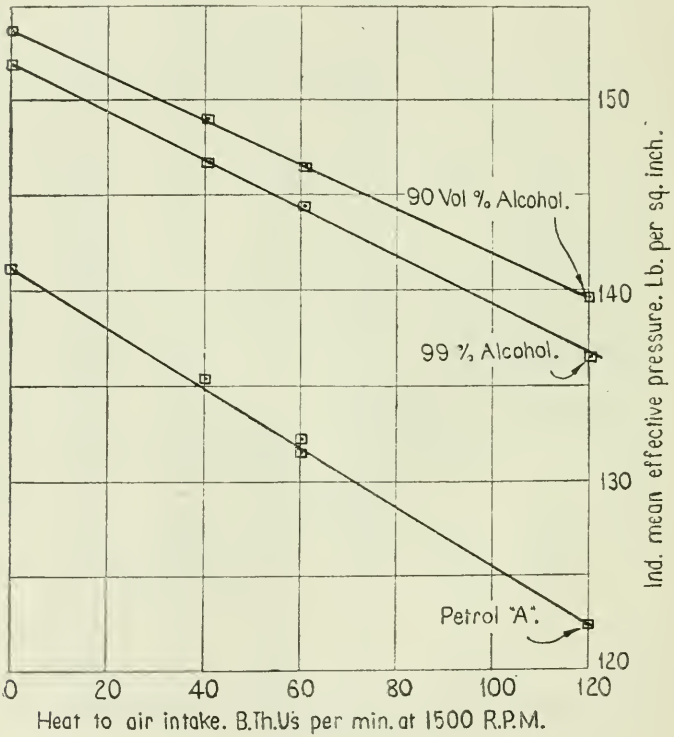
SERIES III.

FIG. 65.—Variation in distribution of heat with mixture strength.

Compression-ratio 5 : 1 ; heating to air intake 60 B.Th.Us. per min.
Alcohol 99 Vol. per cent.
Test 3.

effective pressure with additional water is about the same at the high or low compression, yet when the mixture is enriched beyond this, again there is some divergence, for, at the low compression, the further increase in mean effective pressure with additional water is less than at the higher ratio, suggesting that under these extreme

conditions the heat of compression is barely sufficient at the lower ratio to overcome the latent heat of evaporation of the whole of the liquid. The point is of academic interest only, for the difference is trifling and it is found only when operating at a mixture strength far richer than would ever be employed in practice.



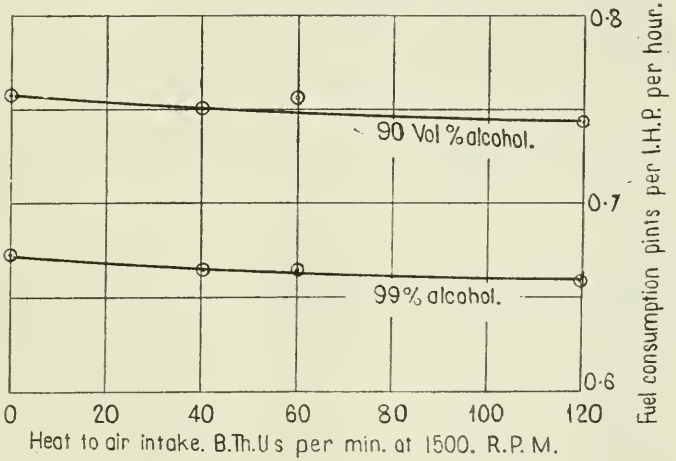
Series III

Fig. 66.—Variation of I.M.E.P. with heating to air intake.

Compression-ratio 5 : 1.
Alcohol 90 and 99 Vol. per cent.
Tests 1–4.

From the test sheets (Series I. and II.) and the graphs Figs. 54 to 57 it will be observed that while at the high compression-ratio the alcohol is burnt at substantially the same thermal efficiency irrespective of water content, at the low compression-ratio the efficiency falls as the water content is increased. This apparent

discrepancy has been observed more than once ; in fact, the tests at the low compression-ratio were repeated three times in order to make absolutely certain that there was no experimental error. It will be observed also that at the low compression-ratio the range of burning on the weak side is narrower than that at the high compression-ratio, but there was no evidence that the water content played any part in this ; in fact, a slightly wider range was actually obtained with the 90 Vol. per cent alcohol than with either of the others, though the difference is very small and of little or no account.



SERIES III.

FIG. 67.—Variation of minimum consumption with heat to air intake.

Compression-ratio 5 : 1.
Alcohol 90 and 99 Vol. per cent.
Tests 1–4.

Generally speaking, the results of this series of tests show that, so long as there are no distribution troubles, the high water content is, on the whole, of little or no disadvantage, for while the efficiency falls (as between the 99 per cent and the 90 Vol. per cent) from 27.5 per cent to 26.6 per cent at a compression-ratio of 3.8 : 1, a drop of 3.3 per cent, at a ratio of 5 : 1 (see next series) it falls only from 32.2 per cent to 31.6 per cent, a drop of less than 2 per cent, and at a ratio of 6 : 1 it does not fall at all. On the other hand, the presence of water in all cases both increases the maximum power output obtainable and reduces the heat-flow through the cylinder walls. It would appear, therefore, that in high-compression engines

there is a substantial advantage to be gained by the presence of additional water and with no loss of efficiency, while at very low compressions, despite the loss in thermal efficiency, the balance of advantage probably lies with the fuel containing the greater quantity of water in solution.

The results of the tests made for Series I. and II. are summarised in Table XV.

TABLE XV.

Nominal strength of fuel. Per cent.	Effective calorific value. B.Th.Us. per pint.	Latent heat of evaporation. B.Th.Us. per pint.	Minimum consumption. Pints per I.H.P. per hour. R = 3.8 : 1.	Minimum consumption. R = 6 : 1.
90 Vol.	10,810	492	0.887	0.655
95 Vol.	11,340	449	0.837	0.632
99	11,890	398	0.788	0.590

Nominal strength of fuel. Per cent.	Thermal efficiency. R = 3.8 : 1. Per cent.	Thermal efficiency. R = 6 : 1. Per cent.	Indicated mean pressure (20 per cent rich). R = 3.8 : 1.	Indicated mean pressure (20 per cent rich). R = 6 : 1.
90 Vol.	26.6	36.2	132.1	157.1
95 Vol.	27.0	35.9	130.8	155.5
99	27.5	36.2	129.5	153.7

Series III.

In this series of tests both the speed and compression were maintained constant at 1500 revs. per minute and 5.0 : 1 respectively;

TABLE XVI.

Heat input. B.Th.Us. per minute.	90 per cent ethyl alcohol.		99 per cent ethyl alcohol.	
	I.M.E.P.	Pints per I.H.P. per hour.	I.M.E.P.	Pints per I.H.P. per hour.
0	152.5	0.763	149.5	0.675
40	148.6	0.752	145.5	0.665
60	145.5	0.753	142.8	0.665
120	137.5	0.745	133.5	0.662

the only variable was the rate of heat-input to the carburettor. Four complete sets of tests were run on each of the two alcohol samples with heat-inputs of 0, 40, 60 and 120 B.Th.Us. per minute. The results obtained are as shown in the accompanying test sheets (Series III., Tests 1–4), pp. 237–240, and graphs Figs. 58–67. The general results may be tabulated as in Table XVI.

In this Table the fuel consumption is that recorded at the most economical mixture strength in each case, while the mean effective pressure is that obtained at a fuel consumption 20 per cent in excess of the most economical.

From these results the following observations may be drawn :—

(1) That the gain in economy with added heat is somewhat greater in the case of 90 Vol. per cent than of 99 per cent alcohol.

(2) That the drop in power with increase of heat is slightly less in the case of 90 Vol. per cent than it is with the 99 per cent alcohol.

(3) That in cases where the problems of uniform distribution do not arise the loss in power, due to the addition of heat, is so large as compared with the small gain in economy that it would appear desirable to use as little pre-heating as possible, more particularly with the higher concentrations of alcohol.

(4) That as the water content increases preheating becomes less objectionable.

The prevalent idea that alcohol requires a great deal of pre-heating and that it should in fact be treated like kerosene is clearly quite erroneous ; only such preheating should be applied as is absolutely necessary to correct errors in distribution.

In the two sets of tests with a heat-input of 60 B.Th.Us. per minute we also made measurements of the distribution of heat-losses and, at the same time, we brought into use our new air-measuring device described on page 213 *ante*. So far as the distribution of heat is concerned the results obtained are shown in the heat-balance sheets and the graphs Figs. 41 and 43, from which it will be observed that at the mixture strength giving complete combustion with each concentration of alcohol the heat-balance is as follows :—

	90 Vol. per cent alcohol. Per cent.	99 per cent alcohol. Per cent.
Heat to I.H.P.	30.9	31.3
Heat to cooling water	23.5	25.0
Heat to exhaust radiation, etc.	45.6	43.7
Total	100.0	100.0

With the larger water content the thermal efficiency is slightly lower and the exhaust losses slightly higher. On the other hand, the heat taken up by the cooling water is distinctly less, indicating that while the temperature of the cycle generally is lower, combustion is less complete or is delayed due to the presence of the water. The air measurements proved particularly interesting. Though expressed in terms of volumetric efficiency, it must be understood that these measurements relate to the volume at n.t.p. of air only entering the cylinder. The air is measured by volume and is reduced to terms of standard pressure and temperature, *i.e.*, 760 mm. mercury and 0° C. It will be observed that when the mixture strength is that giving exactly complete combustion, the mean effective pressure is exactly the same in both cases, namely, 136 lb. per sq. in. With the greater water content the weight of air with this mixture strength is slightly greater, but the increase in power output which might be expected is just counteracted by the drop in thermal efficiency. In the former case 75.5 per cent of air is burnt at an efficiency of 30.7 per cent, and in the latter 74 per cent of air at an efficiency of 31.3 per cent, and it is interesting to note that the product of these two is almost exactly the same. It is also worthy of note that maximum thermal efficiency is obtained when the mixture is approximately 10 per cent weak in the case of 90 Vol. per cent alcohol and 14 per cent weak in the case of the 99 per cent alcohol.

SERIES I. TEST NO. 1.

Power Output and Efficiency.

Compression-Ratio 6 : 1.

Air Standard Efficiency 51.1 per cent.

Fuel—**Ethyl Alcohol 99 per cent.**

S.G. 0.7955. *Mean Temp.* 21°C. *Barometer* 30.08in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input Volts on heater
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
			Petrol "A," compression-ratio 4.6 : 1								
1500	9.9	30.1	126.0	30.4	58.0	30.0	20.5	3.0	57.5	0.515	87*
1506	11.0	29.0	122.0	29.5	67.0	31.0	20.5	3.0	61.9	0.493	87
			99% alcohol compression-ratio 7.0 : 1								
1512	0.3‡	39.3‡	158.0	39.0	57.0	18.0	22.0	3.0	26.9	0.858	87
1496	1.2	38.8	156.0	38.2	58.0	18.5	22.0	3.0	28.7	0.821	87†
			Compression altered to 6.0 : 1								
1496	2.0	38.0	154.5	37.2	57.0	19.0	21.0	3.2	25.4	0.953	87
1504	2.2	37.8	153.8	37.2	59.0	19.0	21.0	3.2	28.8	0.840	86
1496	2.8	37.2	152.0	36.4	62.0	19.0	21.0	3.4	36.3	0.683	86
1500	3.8	36.2	148.5	35.7	65.0	19.3	21.0	3.6	40.1	0.627	87
1512	4.8	35.2	144.5	35.3	63.0	19.5	21.2	3.9	41.1	0.615	87
1496	6.5	33.5	138.5	33.4	61.5	19.5	21.2	4.2	44.8	0.600	86
1504	8.2	31.8	132.5	32.3	60.0	19.7	21.4	4.5	46.4	0.597	87
1516	10.1	29.9	126.0	30.7	58.5	19.7	21.5	4.9	49.6	0.592	87
1500	13.0	27.0	116.0	27.8	56.0	19.5	21.5	5.0	52.2	0.620	87
			Check readings with same fuel and compression								
1500	2.1	37.9	154.0	37.2	57.5	19.0	21.5	3.2	25.4	0.952	87
1500	3.9	36.1	148.0	35.6	61.0	19.3	21.7	3.6	40.5	0.624	87

* = 65 B.Th.Us. per min. † Pre-ignition developed. ‡ Evident error.

REMARKS.—After running for a short period with a compression-ratio of 7 : 1, violent pre-ignition took place. This was probably due to the presence of scale subsequently found on one of the sparking-plugs, but in view of the violence of the pre-ignition, it was deemed advisable to drop the compression-ratio at once to 6 : 1.

SERIES I. TEST NO. 2.

Power Output and Efficiency.

Compression-Ratio 6 : 1.

Air Standard Efficiency 51.1 per cent.

Fuel—**Ethyl Alcohol 90 Vol. per cent.**

S.G. 0.8315. *Mean Temp.* 22°C. *Barometer* 30.08in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	Volts on heater
1	2	3	4	5	6	7	8	9	10	11	12
1496	1.1	38.9	158.0	38.0	56.0	19.0	21.5	3.1	24.8	0.954	87*
1498	1.2	38.8	157.5	38.0	57.5	19.1	21.6	3.1	27.1	0.874	87
1504	1.7	38.3	156.0	37.7	58.5	19.2	21.5	3.1	28.8	0.827	87
1496	2.1	37.9	154.3	37.2	61.0	19.6	21.5	3.5	31.0	0.781	87
1504	2.5	37.5	152.5	37.0	62.0	19.9	21.5	3.6	32.9	0.740	86
1504	2.9	37.1	151.2	36.5	63.0	20.0	21.3	3.8	34.6	0.713	87
1500	4.6	35.4	145.6	35.1	63.0	20.0	21.3	4.0	38.1	0.674	87
1500	6.8	33.2	138.0	33.2	61.0	20.0	21.3	4.2	41.2	0.657	87
1502	9.2	30.8	131.2	31.2	60.0	19.9	21.3	4.5	43.8	0.658	87
1512	10.0	30.0	126.0	30.6	58.0	19.7	21.3	4.9	43.8	0.669	87†
1494	12.4	27.6	117.5	28.3	58.0	19.7	21.3	4.9	45.0	0.705	86‡
		Check readings with same fuel and compression									
1480	1.0	39.0	158.4	37.8	57.5	19.7	21.4	3.2	23.6	1.008	86
1475	1.2	38.8	157.5	37.4	59.5	19.2	21.4	3.2	27.6	0.871	88
1496	3.0	37.0	151.0	36.3	61.5	19.9	21.4	3.7	35.0	0.707	88

* = 65 B.Th.Us. per min. † Ignition too early. ‡ Unstable.

SERIES I. CHECK TEST.

Power Output and Efficiency.

Compression-Ratio 6 : 1.

Air Standard Efficiency 51.1 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

Mean Temp. 20°C. *Barometer* 29.4in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	Volts on heater
1	2	3	4	5	6	7	8	9	10	11	12
1496	1.8	38.2	156.2	37.4	57.0	18.2	20.4	3.3	26.2	0.92	87*
1496	2.0	38.0	155.8	37.2	57.5	18.0	20.4	3.3	28.5	0.848	87
1504	2.5	37.5	154.2	36.8	59.0	18.0	20.4	3.7	31.7	0.772	87
1510	2.7	37.3	153.5	36.7	60.0	18.0	20.4	3.8	33.4	0.734	87
1502	3.5	36.5	150.5	36.0	62.0	18.0	20.2	4.0	36.1	0.683	87
1500	4.2	35.8	148.0	35.3	62.5	18.0	20.2	4.3	38.2	0.668	87
1502	5.4	34.6	143.6	34.3	62.0	18.0	20.2	4.5	40.4	0.648	87
1496	6.6	33.4	139.0	33.4	61.0	18.0	20.2	4.7	42.4	0.635	87
1502	9.0	31.0	130.6	31.4	59.0	18.0	20.3	5.0	44.8	0.637	87
1500	11.2	28.8	122.5	29.3	56.0	18.0	20.3	5.0	47.5	0.647	87
		Check readings on same fuel and compression									
1504	5.5	34.5	143.3	34.2	57.5	18.0	20.2	4.4	40.0	0.658	87
1500	3.2	36.8	151.0	35.9	58.0	18.2	20.4	4.2	32.1	0.780	87

* = 65 B.Th.Us. per min.

REMARKS.—In the last check test, taken rather hurriedly due to shortage of fuel, the ignition was far too advanced; hence the reduced power output and efficiency. It should have been brought back to 3.7 or 3.8.

SERIES II. TEST NO. 1.

Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Ethyl Alcohol 99 per cent.**

S.G. 0.7955. *Mean Temp.* 23°C. *Barometer* 30.41in.

Ignition Position No. 3 = 33 deg. early.

Ignition Position No. 5 = 46 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input Volts on heater
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
1505	7.7	32.3	130.5	32.0	56.0	18.0	21.9	3.3	24.9	1.130	87
1490	8.2	31.8	128.7	31.3	59.0	18.5	22.0	3.6	29.8	0.965	87
1490	9.0	31.0	125.8	30.6	62.8	19.0	22.0	4.0	34.8	0.845	86
1493	9.8	30.2	123.0	30.0	62.0	19.0	22.0	4.4	37.3	0.805	87
1492	11.6	28.4	116.7	28.4	61.0	19.0	21.7	5.0	40.2	0.788	87
1498	13.2	26.8	111.1	27.1	60.0	19.0	22.0	5.2	42.1	0.789	87
1491	13.5	26.5	110.0	26.8	58.0	19.0	22.0	5.5	42.8	0.785	86
1503	10.1	29.9	122.0	29.9	60.8	19.0	22.0	4.6	38.0	0.793	87
1500	8.4	31.6	127.9	31.3	69.0	19.0	22.0	3.8	32.5	0.885	87
1500	7.6	32.4	130.8	32.0	60.8	18.2	21.7	3.4	25.0	1.125	87

REMARKS.—Motoring test : 4.9 lb. on torque arm, 1500 r.p.m., 59.5° C. temp. cooling water.

SERIES II. TEST NO. 2.

Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Ethyl Alcohol 90 Vol. per cent.**

S.G. 0.8315. *Mean Temp.* 20°C. *Barometer* 30.37in.

Ignition Position No. 3 = 33 deg. early.

Ignition Position No. 5 = 46 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input Volts on heater
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
1483	7.3	32.7	132.1	31.9	58.2	15.8	16.0	3.6	22.5	1.255	87
1492	7.2	32.8	132.5	32.2	60.2	15.9	16.0	3.6	23.0	1.215	86.5
1484	7.4	32.6	131.8	31.9	59.0	16.0	16.5	3.8	25.4	1.111	87
1476	7.5	32.5	131.4	31.6	60.2	16.2	16.5	3.8	27.0	1.055	87
1482	7.7	32.3	130.7	31.6	62.1	16.2	16.3	3.8	28.5	1.000	87
1494	7.9	32.1	130.0	31.7	65.0	16.6	16.2	4.1	29.4	0.966	87
1501	8.8	31.2	126.6	30.9	55.2	16.8	17.2	4.3	32.5	0.896	87
1500	8.9	31.1	126.2	30.8	59.3	16.8	17.5	4.3	32.6	0.896	87.5
1494	12.3	27.7	114.3	27.8	58.2	16.1	18.0	5.0	36.3	0.892	86
1506	14.0	26.0	107.9	26.5	58.2	16.0	18.0	5.4	37.3	0.910	87*
1494	9.8	30.2	122.7	29.9	62.1	16.1	18.0	4.6	33.9	0.888	87
1499	7.9	32.1	129.3	31.6	60.4	16.2	18.1	4.1	30.9	0.922	87

* Slightly unstable.

REMARKS.—Motoring test : 4.8 lb., 1500 r.p.m., 60° C. temp. cooling water.

SERIES II. CHECK TEST.

Power Output and Efficiency.

Compression-Ratio 3.8 : 1.

Air Standard Efficiency 41.4 per cent.

Fuel—**Ethyl Alcohol 95 Vol. per cent.**

S.G. 0.8155. *Mean Temp.* 22°C. *Barometer* 30.41in.

Ignition Position No. 3 = 33 deg. early.

Ignition Position No. 5 = 46 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input Volts on heater
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
				Preliminary run on Petrol "A"							
1500	11.5	28.5	117.5	28.7	63.0	29.5	20.0	3.0	54.9	0.572	88
1500	11.5	28.5	117.5	28.7	65.0	29.5	20.5	3.0	50.8	0.617	88.5
		Changed fuel (without stopping) to alcohol									
1500	7.8	32.2	130.5	31.9	60.5	18.2	20.7	3.5	26.1	1.08	87
1489	8.1	31.9	129.5	31.4	62.6	18.5	21.5	3.5	29.5	0.973	87
1501	9.6	30.4	124.1	30.4	63.9	18.3	21.5	4.0	34.8	0.850	86.5
1488	10.0	30.0	122.7	29.8	58.5	18.2	21.5	4.2	35.8	0.844	86
1486	13.3	26.7	111.2	27.0	61.5	18.5	21.9	5.0	39.5	0.844	87
1502	13.4	26.6	110.8	27.2	59.2	18.2	21.7	5.5	39.4	0.840	87*
1490	11.1	28.9	119.0	28.9	61.2	18.5	21.8	5.0	37.3	0.836	87
1495	9.0	31.0	126.3	30.5	65.0	19.0	21.5	4.6	33.7	0.876	87
1497	7.6	32.4	131.2	32.0	57.0	18.0	21.6	3.3	24.5	1.15	87

* Slightly unstable.

Test could not be extended owing to the last of this alcohol 95 Vol. per cent having been used.

REMARKS.—Motoring test : 5.0 lb. on torque arm, 1500 r.p.m., 58° C. temp. of cooling water.

SERIES III. TEST NO. 1.

Effect of Varying Heating on Power Output and Efficiency.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 90 Vol. per cent.**

S.G. 0.8315. *Mean Temp.* 19°C. *Barometer* 29.83in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	No heat
1	2	3	4	5	6	7	8	9	10	11	12
					Petrol	"A"					Volts
1500	5.3	34.7	141.2	34.0	62.0	13.0	19.0	3.5	50.2	0.528	0 N.D.
		Ran on from Petrol "A" to 90 Vol. per cent. alcohol									Volts
1500	1.8	38.2	153.5	37.0	58.0	10.8	19.0	3.5	24.0	1.014	0
1486	2.0	38.0	153.0	36.5	60.5	11.1		3.5	26.0	0.948	,,
1502	2.5	37.5	151.1	36.4	59.5	11.8	20.0	3.5	28.6	0.864	,,
1502	3.5	36.5	147.7	35.6	60.5	12.0		3.5	31.8	0.795	,,
1487	4.4	35.6	144.4	34.5	61.0	12.1		4.5	33.6	0.776	,,
1511	4.3	35.7	144.9	35.1	61.0	12.1	20.1	4.5	33.2	0.772	,,
1504	5.9	34.1	139.1	33.6	59.5	12.0		5.0	35.6	0.752	*
1516	6.9	33.1	135.5	33.0	58.5	12.0		5.2	35.6	0.756	*
1507	7.0	33.0	135.1	32.7	58.0	11.9	20.1	5.2	36.0	0.765	*
1506	9.4	30.6	126.6	30.6	57.5	11.9		5.5	37.6	0.782	†
1501	9.3	30.7	127.0	30.6	57.0	11.9	20.1	5.8	37.4	0.787	‡
1484	1.8	38.2	153.7	36.6	57.0	12.0		3.5	24.6	1.00	
Ran on from 90 Vol. per cent alcohol to 99 Vol. per cent alcohol§											Volts
1500	2.3	37.7	151.9	36.6	55.0	11.8	20.2	3.5	23.2	1.06	0
1490	2.3	37.7	151.9	36.3	59.5	11.9		3.5	24.0	1.035	,,
1490	2.5	37.5	151.1	36.2	62.0	11.9		3.5	28.2	0.882	,,
1492	2.75	37.25	150.3	36.0	60.0	12.0	20.5	3.8	29.6	0.845	,,
1496	3.3	36.7	148.4	35.6	61.0	11.9		4.2	32.9	0.768	,,
1506	4.6	35.4	143.7	34.7	62.0	11.8		4.7	37.6	0.690	,,
1504	5.9	34.1	139.1	33.6	60.5	11.8	20.7	5.0	39.6	0.677	,,
1498	8.0	32.0	131.6	31.6	58.0	11.7		5.5	42.3	0.674	,,
1506	10.2	29.8	123.8	29.9	55.0	11.6		5.7	44.0	0.684	†
1506	9.9	30.1	124.8	30.2	55.0	11.6	21.0	5.8	44.0	0.677	‡
1500	2.6	37.4	150.9	36.3	58.0	11.8		3.5	28.8	0.861	

* Very slightly unstable. † Unstable. ‡ Slightly unstable.

§ Sp. gr., 0.7955 ; mean temp., 20° C.

SERIES III. TEST NO. 2.

Effect of Varying Heating on Power Output and Efficiency.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 90 Vol. per cent.**

S.G. 0.8315. *Mean Temp.* 22°C. *Barometer* 29.9in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ‡ pint.		Heat input Volts on heater
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
					Petrol "A"						
1500	6.8	33.2	135.5	32.7	59.5	23.5	21.0	3.5	51.2	0.538	68
	90 Vol.	per ce	nt. ra	n on f	rom te	st on	Petrol	"A"			
1507	3.1	36.9	148.9	35.8	60.0	15.0	20.9	3.5	25.0	1.006	68
1496	3.2	36.8	148.6	35.7	59.6	15.0	20.8	3.5	27.9	0.904	67
1501	3.7	36.3	146.8	35.4	60.0	15.1		3.5	30.7	0.828	68
1498	4.4	35.6	144.3	34.7	60.5	15.3		3.5	33.0	0.786	70
1500	5.4	34.6	140.7	33.9	61.0	15.5	20.5	4.0	34.4	0.772	68
1501	6.9	33.1	135.4	32.7	59.0	15.4		4.3	36.3	0.758	67
1506	8.9	31.1	128.2	31.1	57.0	15.3		5.0	38.4	0.754	68
1513	9.8	30.2	125.1	30.4	56.0	15.2	20.6	5.3	39.5	0.750	68*
1502	11.6	28.4	118.7	28.7	57.0	15.0		5.8	40.9	0.767	67†
			Ran	on wi	th 99	Vol.	per ce	nt al	cohol‡		
1505	3.6	36.4	147.1	35.6	61.0	14.8	20.7	3.5	26.0	0.978	67
1546	3.85	36.15	146.2	36.3	60.5	15.0		3.5	28.4	0.873	68
1494	4.1	35.9	145.4	34.9	61.5	15.0		3.5	32.6	0.791	68
1489	5.5	34.5	140.4	33.6	61.3	15.1	20.7	4.0	38.2	0.701	69
1505	6.6	33.4	136.4	33.0	61.5	15.2		4.4	39.4	0.692	67
1503	8.3	31.7	130.4	31.5	58.0	15.2	20.7	5.1	42.8	0.668	69
1509	9.6	30.4	125.8	30.5	57.0	15.0		5.4	44.4	0.665	68
1498	11.3	28.7	119.7	28.8	57.0	14.7		5.7	46.8	0.668	67
1506	13.6	26.4	111.5	27.0	56.0	14.6	20.7	5.8	48.6	0.686	68*

* Slightly unstable. † Unstable.

‡ Sp. gr., 0.7955 ; mean temp., 22° C.

SERIES III. TEST NO. 3.

Effect of Varying Heating on Power Output and Efficiency.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 90 Vol. per cent.**

S.G. 0.8315. *Mean Temp.* 22°C. *Barometer* 29.99in.

Ignition Position No. 3 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M.E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input Volts on heater
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
					Petrol "A"						
1509	7.7	32.3	132.2	32.1	60.8	29.0	20.0	3.5	51.4	0.546	84
1501	7.9	32.1	131.5	31.9	61.6	30.2	20.0	3.5	53.1	0.532	87
		Ran on from Petrol "A" to 90 Vol. per cent alcohol									
1496	4.0	36.0	145.3	35.0	58.8	17.5	20.0	3.5	26.3	0.978	85*
1487	4.4	35.6	143.9	34.5	61.0	18.6	20.2	3.5	31.0	0.842	85
1484	5.25	34.75	140.9	33.7	63.3	19.0		4.0	33.8	0.790	84
1492	6.5	33.5	136.5	32.8	58.0	19.0	20.4	4.2	35.7	0.768	82
1491	7.9	32.1	131.5	31.6	56.0	18.5		4.8	37.5	0.760	80
1497	9.1	30.9	127.2	30.7	59.7	18.8	20.4	5.3	38.8	0.756	84
1500	10.3	29.7	123.0	29.7	58.5	19.0		5.5	39.7	0.763	84
1502	11.5	28.5	118.7	28.75	57.3	19.3	20.4	5.8	40.6	0.772	85
1504	3.3	36.7	147.8	35.8	53.0	18.3		3.6	22.5	1.117	85
1486	3.3	36.7	147.8	35.4	53.3	18.2	20.5	3.6	23.1	1.100	85
1471	3.9	36.1	145.7	34.6	56.6	18.6		3.6	28.4	0.916	85
			Ran on with 99 Vol. per cent alcohol†								
1490	4.2	35.8	145.6	34.7	57.3	18.0	20.6	3.6	23.2	1.118	85*
1482	4.3	35.7	144.3	34.5	58.6	18.1		3.6	23.0	1.135	85
1480	4.3	35.7	144.3	34.4	61.8	18.4	20.7	3.6	28.7	0.912	85
1486	4.7	35.3	142.9	34.2	59.6	19.0		3.6	32.3	0.815	84
1489	5.5	34.5	140.0	33.6	61.9	19.1	20.8	3.9	38.2	0.702	85
1504	6.3	33.7	137.2	33.3	62.2	19.1	20.9	4.4	39.1	0.692	85
1511	7.6	32.4	132.6	32.3	60.5	19.1	20.9	4.6	41.3	0.675	85
1488	8.9	31.1	128.0	30.7	58.5	19.1		5.0	43.7	0.671	85
1501	9.4	30.6	126.2	30.5	57.0	19.0		5.1	44.3	0.666	84
1506	11.0	29.0	120.5	29.3	56.0	19.0	21.0	5.3	46.4	0.662	85
1510	12.4	27.6	115.5	28.1	57.0	19.1		5.5	47.2	0.679	85
1495	14.5	25.5	108.0	26.0	57.0	19.0		5.8	49.3	0.702	84
1503	12.5	27.5	115.1	27.9	57.2	19.1	21.0	5.8	47.6	0.678	84
1502	4.2	35.8	144.7	35.0	59.2	18.5		3.6	26.2	0.982	85

* = 60 B.Th.Us. per min.

† Sp. gr., 0.7955 ; mean temp., 23° C.

SERIES III. TEST No. 4.

Effect of Varying Heating on Power Output and Efficiency.

Compression-Ratio 5 : 1.

Air Standard Efficiency 47.47 per cent.

Fuel—**Ethyl Alcohol 90 Vol. per cent.**

S.G. 0.8315. *Mean Temp.* 26°C. *Barometer* 30.06in.

Ignition Position No. 3 = 24 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input Volts on heater
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
					Petrol	"A"					
1500	10.4	29.6	122.4	29.6	61.5	45.5	25.0	3.5	54.5	0.558	115*
		Ran on f	rom	Petro	l "A"	to 90	Vol. p	er cen	t alcoh	ol	
1492	5.5	34.5	139.6	33.6	55.0	22.5	25.5	3.5	24.6	1.09	115†
1471	5.9	34.1	138.2	32.9	62.0	22.7		3.5	29.2	0.937	114
1491	7.2	32.8	133.7	32.2	61.0	23.0	25.8	3.5	33.8	0.827	114
1499	8.0	32.0	130.8	31.7		23.0		4.0	37.0	0.768	114
1501	8.6	31.4	128.7	31.2		23.1		4.5	37.9	0.762	115
1500	9.6	30.4	125.1	30.3	60.0	23.1	26.0	4.5	39.5	0.752	115
1501	11.0	29.0	120.1	29.1	59.0	23.1		5.2	41.4	0.747	115
1494	12.4	27.6	115.2	27.8	57.0	23.0		5.7	43.7	0.741	115
1505	14.4	25.6	108.1	26.3	55.0	23.0	26.1	5.8	45.3	0.755	115‡
1486	5.6	34.4	139.4	33.4	60.0	22.8		3.5	24.7	1.091	115
			Ran	on to	99 Vol.	per c	ent	al	cohol	‖	
1489	6.8	33.2	135.1	32.5	57.0	22.5	26.5	3.5	23.7	1.169	115
1497	6.6	33.4	135.8	32.8		22.2		4.0	23.4	1.173	115
1482	7.0	33.0	134.4	32.2	56.0	22.0	27.0	4.0	23.0	1.216	114§
1489	6.3	33.7	136.9	32.9	60.0	22.5		3.5	25.3	1.081	115
1469	6.4	33.6	136.5	32.4	61.0	22.5		3.5	26.8	1.037	114
1476	6.7	33.3	135.4	32.3	63.0	23.0	27.6	3.6	29.1	0.958	115
1500	7.7	32.3	131.9	32.0	61.0	23.5		4.0	37.0	0.760	115
1497	8.6	31.4	128.7	31.1	63.0	23.4	28.3	4.4	40.4	0.716	115
1504	10.1	29.9	123.4	30.0	60.0	23.2	28.5	4.9	43.9	0.683	114
1500	11.6	28.4	118.1	28.6	60.0	23.2	28.7	5.2	46.7	0.674	115
1498	13.25	26.75	112.2	27.2	58.0	23.0	29.0	5.5	50.0	0.662	114
1504	14.9	25.1	106.3	25.9	56.0	23.0	29.0	5.8	53.0	0.656	114‡

* V.O.D. † = 120 B.Th.Us. per min. ‡ Just stable. § Unstable.

‖ Sp. gr., 0.7955 ; mean temp., 27° C.

HEAT-BALANCE SHEET. SERIES III. TEST NO. 3.

Test on Ethyl Alcohol 90 Vol. per cent. *Compression-Ratio* 5 : 1.
Heat to Air Intake, 60 B.Th.Us. per minute.

I.M.E.P. lb. per sq. in.	Heat in B.Th.Us. per hour. Total heat of fuel supplied	Heat to I.H.P.	Heat to cooling water	Distribution of heat, per cent. Heat to I.H.P.	Heat to cooling water	Heat to exhaust radiation, etc.
B.S.R.						
132.2	317,500	81,700	64,550	25.7	20.4	53.9
131.5	307,000	81,200	67,150	26.4	21.8	51.8
Alcohol						
145.3	368,000	89,050	56,000	24.2	15.2	60.6
143.9	312,000	87,800	59,700	28.1	19.1	52.8
140.9	286,000	85,750	62,450	30.0	21.8	48.2
136.5	271,000	83,450	64,850	30.8	23.9	45.3
131.5	258,000	80,400	61,500	31.1	23.8	45.1
127.2	249,000	78,100	58,600	31.4	23.5	45.1
123.2	243,500	75,550	56,700	31.1	23.3	45.6
118.7	238,000	73,200	55,200	30.8	23.2	46.0
147.8	430,000	91,100	57,700	21.2	13.4	65.4
147.8	418,500	90,100	52,850	21.6	12.7	65.7
145.7	340,500	88,050	58,000	25.8	17.0	57.2

VOLUMETRIC EFFICIENCY. SERIES III. TEST NO. 3.

Test on Ethyl Alcohol 90 Vol. per cent. *Compression-Ratio* 5 : 1.
Heat to Air Intake, 60 B.Th.Us. per minute.

I.M.E.P.	Fuel consumption (pints per I.H.P. per hour)	* Volumetric efficiency	Air/fuel ratio by weight
132.2	0.546	72.4	10.77
131.5	0.532	71.9	10.99
145.3	0.978	77.2	5.82
143.9	0.842	76.3	6.74
140.9	0.790	75.6	7.26
136.5	0.768	75.7	7.72
131.5	0.760	75.9	8.13
127.2	0.756	75.8	8.42
123.0	0.763	76.3	8.71
118.7	0.772	76.7	8.94
147.8	1.117	78.9	5.15
147.8	1.100	77.7	5.11
145.7	0.916	76.8	6.15

* Volumetric efficiency is here defined as the ratio (per cent) of the weight of air passed through the engine per cycle to weight of the swept volume of air at 0° C.

HEAT-BALANCE SHEET. SERIES III. TEST No. 3.

Test on Ethyl Alcohol 99 Vol. per cent. *Compression-Ratio* 5 : 1
Heat to Air Intake, 60 B.Th.Us. per minute.

I.M.E.P.	Heat in B.Th.Us. per hour			Distribution of heat, per cent.		
lb. per sq. in.	Total heat of fuel supplied	Heat to I.H P.	Heat to cooling water	Heat to I.H.P.	Heat to cooling water	Heat to exhaust radiation, etc.
145.6	458,000	88,250	54,550	19.3	11.9	68.8
144.3	462,000	87,750	55,150	19.0	11.9	69.1
144.3	370,500	87,500	59,650	23.6	16.1	60.3
142.9	329,000	87,000	64,350	26.4	19.5	54.1
140.0	278,500	85,500	67,000	30.7	24.0	45.3
137.2	272,000	84,700	68,500	31.1	25.2	43.7
132.6	257,500	82,150	65,800	31.9	25.6	42.5
128.0	243,000	78,100	62,700	32.1	25.8	42.1
126.2	240,000	77,600	61,000	32.3	25.4	42.3
120.5	229,000	74,550	59,400	32.5	25.9	41.6
115.5	225,000	71,450	53,700	31.7	23.9	44.4
108.0	215,500	66,150	53,700	30.7	24.9	44.4
115.1	223,500	71,000	53,250	31.8	23.8	44.6
144.7	406,000	89,000	56,600	21.9	13.9	64.2

VOLUMETRIC EFFICIENCY. SERIES III. TEST No. 3.

Test on Ethyl Alcohol 99 Vol. per cent. *Compression-Ratio* 5 : 1.
Heat to Air Intake, 60 B.Th.Us. per minute.

I.M.E.P.	Fuel consumption (pints per I.H.P. per hour)	* Volumetric efficiency	Air/fuel ratio by weight
145.6	1.118	78.9	5.48
144.3	1.135	79.1	5.41
144.3	0.912	76.3	6.50
142.9	0.815	75.1	7.23
140.0	0.702	74.0	8.44
137.2	0.692	74.1	8.74
132.6	0.675	73.7	9.23
128.0	0.671	75.0	9.77
126.2	0.666	74.6	9.96
120.5	0.662	74.9	10.50
115.5	0.679	75.6	10.80
108.0	0.702	75.1	11.11
115.1	0.678	75.3	10.80
144.7	0.982	76.1	6.01

* Volumetric efficiency is here defined as the ratio (per cent) of the weight of air passed through the engine per cycle to weight of the swept volume of air at 0° C.

TESTS OF ALCOHOL-ETHER MIXTURES.

General Notes.

The following report has been divided into two sections, the first dealing with tests carried out on a single-cylinder variable-compression engine, and the second with further tests carried out on a four-cylinder Leyland lorry engine.

The fuels used for these tests were as follows :—Ethyl alcohol 95 vol. per cent ex Hammersmith Distillery, ether ex Gretna, and petrol "A," a straight run distillate F.B.P. about 180° C. ex Borneo.

Various data (some measured and some calculated) are given in Table XVII. for these fuels, and the mixtures thereof. as used in the engine tests :—

TABLE XVII.

Fuel or mixture	S.G. at 15° C.	Latent heat, B.Th.Us. per lb.	Effective calorific value	
			B.Th.Us. per lb.	B.Th.Us. per pint
Petrol "A"	0.785	140	18,550	18,200
Alcohol 95 Vol. per cent	0.8155	440	11,130	11,350
Ether, ex Gretna	0.7395	215	13,650	12,600
Alcoh. 95 per cent., ether 5 per cent ..	0.813	428	11,240	11,420
,, 90 ,, ,, 10 ,, ..	0.810	417	11,320	11,470
,, 85 ,, ,, 15 ,, ..	0.8055	406	11,460	11,540

NOTE.—The alcohol-ether mixtures were made by volume.

The ether is assumed to contain 7 per cent of water by weight.

SECTION I.

The experiments described in this section have all been carried out on a new variable-compression engine and new apparatus throughout. This new engine and its equipment is an almost exact

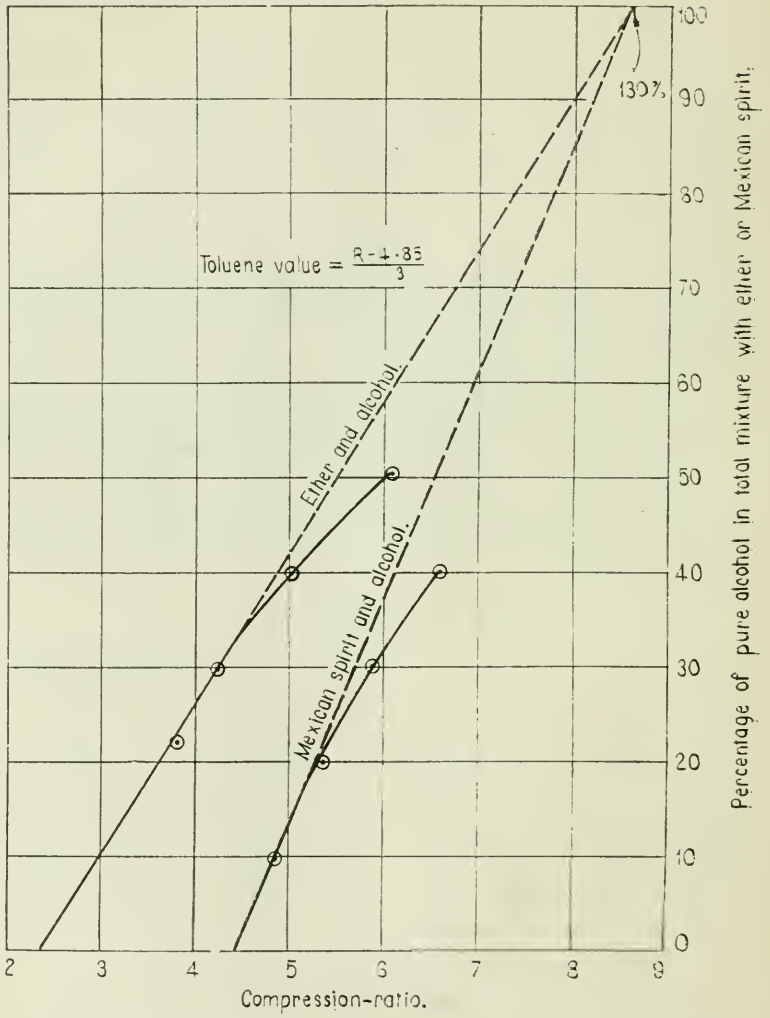


FIG. 68.—Detonation tests on variable-compression engine.

Heating to carburettor, water outlet temperature and speed constant. Full throttle. Mixture strength nearly 10 per cent rich. Ignition adjusted for maximum power at each compression.

replica of the old plant described fully in the first report; it differs only in that, as a result of experience with the older engine, some minor improvements have been made in the way of facilitating handling; also the valve setting has been altered slightly, again as the result of experience with the older unit, to give a higher volumetric efficiency and therefore a slightly higher power output at the same compression-ratio. In all other respects, it is identical with the older unit on which the experiments dealt with in the first and second reports were carried out, and the procedure adopted has been exactly the same.

Programme of Tests on Variable-Compression Single-Cylinder Unit.

The programme consisted of the following series of tests :—

Series I.—Tests to determine the detonation-point of mixtures of alcohol 95 Vol. per cent and ether. Three such tests to be run, the most suitable fuel proportions in order to keep well within the range of compression available to be determined by experiment.

Series VI.—Ascertain the influence of small additions of pure alcohol upon Pratt's petrol as regards detonation.

Series II.—Consumption and power tests over the complete mixture range at constant speed, constant heat input and constant jacket temperature and at a compression-ratio of 5.45 : 1 on three mixtures, consisting of 95 Vol. per cent alcohol with 5, 10 and 15 per cent by weights of ether (three complete tests).

Series III.—Consumption and power tests over complete mixture range at 40 per cent load and full load at constant speed, with heat input to carburettor proportional to the load, and with fixed ignition timing as adjusted for full load on the above mixtures and also on 95 Vol. per cent alcohol alone, with additional check tests on each mixture at 40 per cent load and with ignition adjusted for best economy (twelve complete tests).

Series I. and VI.

Detonation Tests on Mixtures of Alcohol, Ether and Petrol.

It is convenient to consider the results of the above two series of tests together.

For these tests pure alcohol (99.4 per cent) was used, since commercial alcohol (95 Vol. per cent) cannot be mixed with petrol when the latter is in excess.

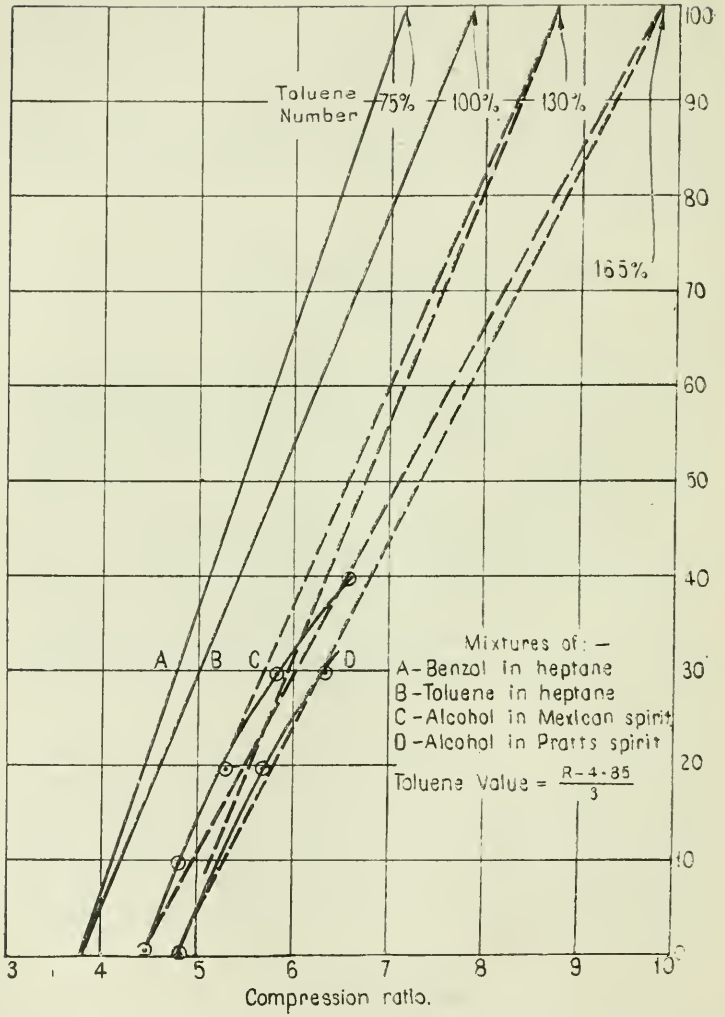


FIG. 69.—Detonation tests on variable-compression engine.

Heating to carburettor, water outlet temperature and speed constant. Full throttle. Mixture strength nearly 10 per cent rich. Ignition adjusted for maximum power at each compression.

The detonation results for mixtures of pure alcohol and ether may safely be taken as sufficiently correct also for mixtures of commercial alcohol and ether.

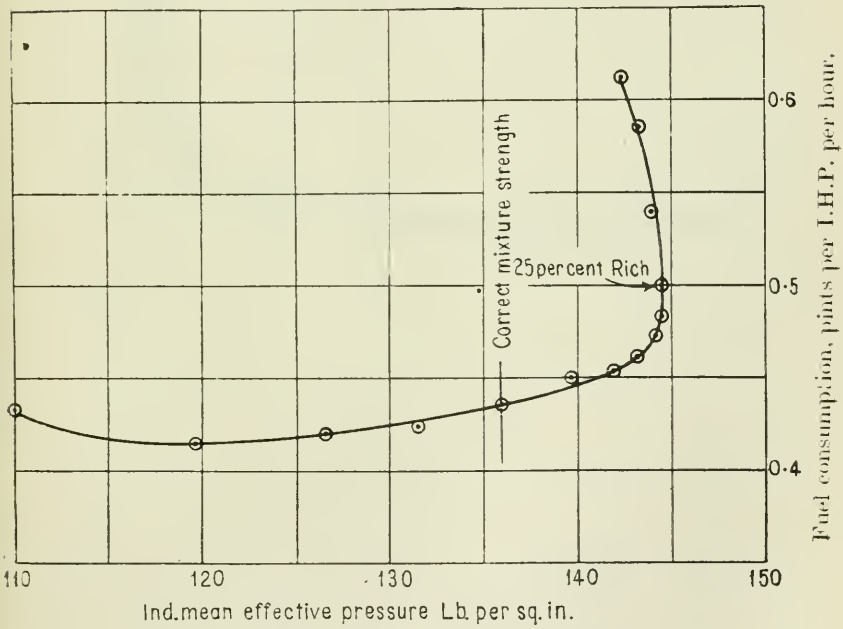
The following mixtures were tested :—

(1) Ether with 22, 30, 40 and 50 per cent of pure alcohol.

(2) Mexican spirit with 0, 10, 20, 30 and 40 per cent of pure alcohol.

(3) Pratt's motor spirit with 0, 10, 20 and 30 per cent of pure alcohol.

In each case the engine was started up at a low compression-ratio, and run for a period at first on the petrol alone and then on mixture



SERIES II.—VARIABLE-COMPRESSION ENGINE.

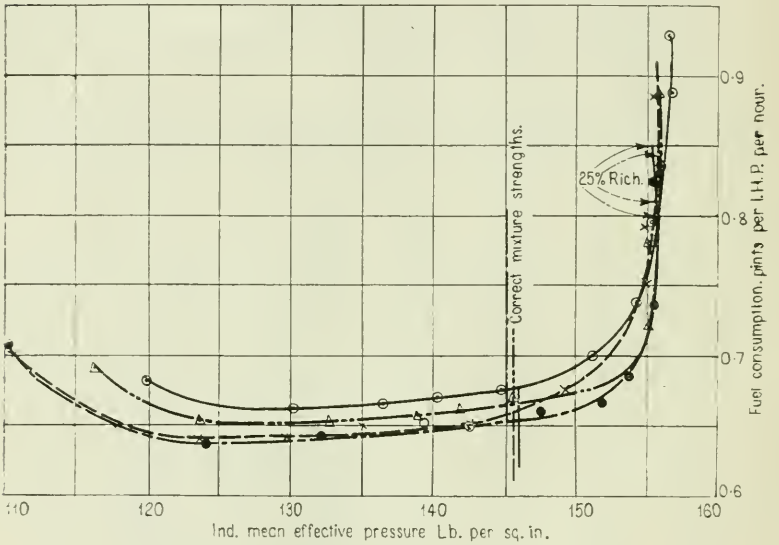
FIG. 70.—Variation of I.M.E.P. and consumption with mixture strength.

Compression-ratio 5.45 : 1 ; full load 1500 revs. per minute ; heating to air intake 65 B.Th.Us. per minute (net) ; ignition adjusted ; jacket water temperature 60° C.

Petrol, A.

Test 1.

until all necessary adjustments had been made and all temperature conditions, etc., had settled down to their normal values. The mixture strength was adjusted to approximately 10 per cent rich at which strength detonation is at its maximum, and the ignition advanced until the torque was at a maximum. The cylinder was



SERIES II.—VARIABLE-COMPRESSION ENGINE.

FIGS. 71, 72, 73, 74.—Variation of I.M.E.P. and consumption with mixture strength.

Compression-ratio 5.45 : 1 ; full load, 1500 revs. per minute ; heating to air intake 65 B.Th.Us. per min. (net) ; ignition adjusted.

———— — Alcohol 95 Vol. per cent, jacket water temperature 52° C. Test 2.
—— - - —— Alcohol and ether 5 per cent, jacket water temperature 52° C. Test 3.
— — — — — Alcohol and ether 10 per cent, jacket water temperature 53° C. Test 4.
—— - — - —— Alcohol and ether 15 per cent, jacket water temperature 51° C. Test 5.

then lowered very gradually until detonation became apparent by a sharp ringing knock in the cylinder, followed, after an interval of a few minutes, by a slight drop in torque, due to incipient pre-ignition.

The exact determination of the detonation-point, as found in this manner, is, to some extent, a question of judgment on the part of

the observer; also the accuracy of determination depends in some measure upon the nature of the fuel. In the case of certain fuels, detonation is very clearly defined; in others, it is much less definite. Both with the ether-alcohol and with the petrol-alcohol mixtures, the detonation-point was very sharply defined, and could be determined with certainty to within ± 0.05 in the compression-ratio, that is to say, while at a compression-ratio of, say, 4.8 : 1 there would be only very slight detonation, at 4.9 : 1 there would be a loud ringing knock.

The results of these tests are shown in the graphs, Figs. 68 and 69.

Conclusions based on Tests under Series I. and VI.

From these graphs the following may be concluded :—

(1) With all mixtures containing alcohol the compression-ratio at the detonation-point increases more rapidly than in direct proportion to the percentage of alcohol.

(2) All tests are in good agreement that for small proportions of alcohol its effective toluene value is 130 per cent, the effect of pure toluene being taken as unity, while that of commercial benzol is 75 per cent.

(3) For mixtures containing between 30 and 40 per cent of alcohol, its effective toluene value may be taken as 160 per cent.

(4) The detonation-points of the various bodies were as follows :—

Ether (determined by extrapolation of the curve)	2.4 : 1	compression-ratio
Heptane	3.8 : 1	,,
Mexican spirit	4.4 : 1	,,
Pratt's motor spirit	4.8 : 1	,,

Series II.

Power and Consumption Tests at Full Load and Ignition Adjusted.

This series comprised a number of fuel-consumption tests on petrol "A," 95 Vol. per cent alcohol, and mixtures containing this alcohol with 5, 10 and 15 per cent of ether respectively. The tests were made with wide-open throttle at a speed of 1500 revs. per minute, at a compression-ratio of 5.45 : 1 and with the mixture strength varied from that giving maximum power down to the weakest which could be burned, and in all cases with a constant heat input of 65 B.Th.Us. per minute and the ignition varied to suit each mixture strength.

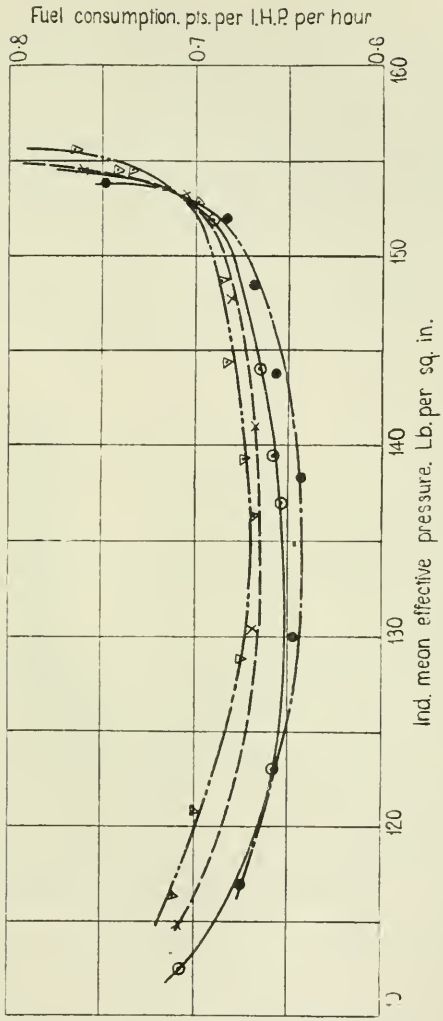
The results of these tests are shown in detail on the test sheets, pp. 270–274, and on the graphs, Figs. 70 to 74.

The significant points on each of these curves are shown in Table XVIII.

TABLE XVIII.—RESULTS WITH VARIABLE-COMPRESSION ENGINE.

Speed 1500 *r.p.m., Compression-Ratio* 5.45 : 1.

Test and Series No.	Fuel	Load throttle-opening	Ignition fixed or adjusted	Heat to air intake, B.Th. Us. per min.	Power I.M.E.P. (lb. per sq. in.)		Fuel consumption (pints per I.H.P. per hour)			Ind. th. efficiency (per cent.)	
					When 25 per cent rich	Correct mixture	25 per cent rich	Correct mixture	Minimum (weak)	Correct mixture	Maximum (weak)
1 Ser. II.	Petrol "A"	Full	Adjusted	65	144.5	136.0	0.50	0.437	0.415	32.0	33.7
2 „ „	Alcohol 95 vol. per cent.	„	„	„	156.0	146.0	0.84	0.680	0.660	33.0	34.0
3 „ „	Alcoh. ether 5 per cent.	„	„	„	155.5	145.5	0.83	0.668	0.651	33.3	34.2
4 „ „	„ „ 10 per cent.	„	„	„	155.5	145.5	0.81	0.660	0.643	33.6	34.5
5 „ „	„ „ 15 per cent.	„	„	„	155.5	145.0	0.80	0.655	0.639	33.7	34.5
1 Ser. III.	Alcohol 95 vol. per cent.	Full	Fixed	65	—	—	—	—	0.670	—	33.5
4 „ „	Alcoh. ether 5 per cent.	„	„	„	—	—	—	—	0.665	—	33.5
7 „ „	„ „ 10 per cent.	„	„	„	—	—	—	—	0.658	—	33.7
10 „ „	„ „ 15 per cent.	„	„	„	—	—	—	—	0.642	—	34.3
3 Ser. III.	Alcohol 95 vol. per cent.	40 per cent.	Adjusted	26	77.0	—	—	—	0.670	—	33.5
6 „ „	Alcoh. ether 5 per cent.	„	„	„	77.0	—	—	—	0.665	—	33.5
9 „ „	„ „ 10 per cent.	„	„	„	77.2	—	—	—	0.645	—	34.4
12 „ „	„ „ 15 per cent.	„	„	„	77.5	—	—	—	0.630	—	35.0
2 Ser. III.	Alcohol 95 vol. per cent.	40 per cent.	Fixed	26	78.0	—	—	—	0.675	—	33.2
5 „ „	Alcoh. ether 5 per cent.	„	„	„	77.5	—	—	—	0.665	—	33.5
8 „ „	„ „ 10 per cent.	„	„	„	78.0	—	—	—	0.655	—	33.9
11 „ „	„ „ 15 per cent.	„	„	„	77.6	—	—	—	0.655	—	33.7



SERIES III.—VARIABLE-COMPRESSION ENGINE.

FIGS. 75, 76, 77 AND 78.—Effect of addition of ether on efficiency.

Compression-ratio 5.45 : 1 ; full load 1500 revs. per minute ; heating to air intake 65 B.Th.Us. per minute : fixed ignition.

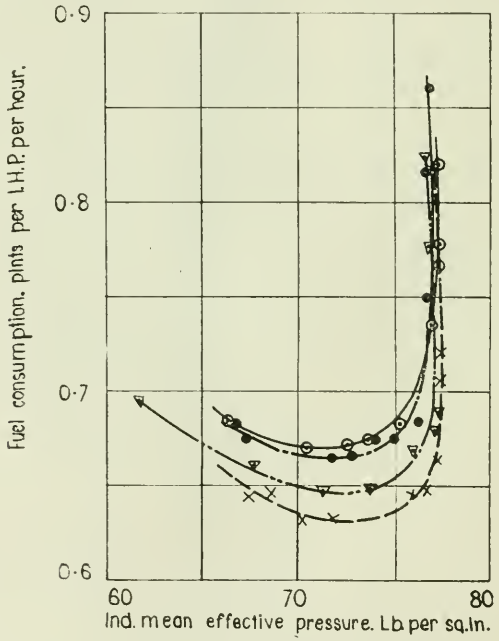
—·—·— 95 Vol. per cent alcohol ; jacket temperature 57° C. Test 1.

— — — — Alcohol and ether 5 per cent ; jacket temperature 54° C. Test 4.

———— Alcohol and ether 10 per cent ; jacket temperature 52° C. Test 7.

— — · — — Alcohol and ether 15 per cent ; jacket temperature 52° C. Test 10.

It will be observed that two extra tests were made under Series II., one with petrol "A," and one with the 95 Vol. per cent alcohol alone. Tests had also been made to determine the volumetric efficiency of this single-cylinder engine with petrol "A" and with alcohol under the same conditions as on the present tests. Although we saw no advantage in recording, in this report, the details of these preliminary tests, we have been able from the information which they provided to show on the graphs the points on the fuel-consumption curves where the mixture strength was "correct"—that is,



SERIES III.—VARIABLE-COMPRESSION ENGINE.

FIGS. 79, 80, 81 AND 82.—Effect on I.M.E.P. and consumption of adding ether to alcohol.

Compression-ratio 5.45 : 1 ; heating to air intake 26 B.Th.Us. per min.; load 40 per cent ; 1500 revs. per minute ; ignition adjusted.

———————— Alcohol 95 Vol. per cent ; jacket temperature 56° C. Test 3.
— — — — — — Alcohol and ether 5 per cent ; jacket temperature 55° C. Test 6.
— — — — — — Alcohol and ether 10 per cent ; jacket temperature 55° C. Test 9.
— — — — — — Alcohol and ether 15 per cent ; jacket temperature 54° C. Test 12.

where the proportion of fuel to air was such as to render possible the complete combustion of both the fuel and the oxygen. It has been possible also to show the point on each curve where the mixture was 25 per cent "rich," and this point has been taken as a basis for comparing the maximum power output, or indicated mean effective pressure, derived from each fuel. As may be seen from Table XVIII., we have given the fuel consumption and efficiency obtained with the "correct" mixture strength as well as with the mixture "weak." The latter mixture in each case is taken as that at which the fuel consumption per indicated horse-power per hour, and therefore the efficiency, is most favourable. Both for petrol, for alcohol and for alcohol-ether mixtures, it was found that maximum efficiency occurred when there was from 15 to 17 per cent of excess air present in the mixture.

Conclusions Based on Tests, Series II.

Under the conditions of these tests :—

(1) Power falls very slightly with increasing proportions of ether. Between the limits of alcohol alone and alcohol with 15 per cent ether, the fall in power, both with "25 per cent rich" and with "correct mixture," is less than 1 per cent.

(2) For both "25 per cent rich" and for "correct mixture," the power output derived from alcohol is between 8 and 7.5 per cent greater than that given by petrol.

(3) Efficiency increases with increasing proportions of ether. Between the limits of alcohol alone and alcohol with 15 per cent ether, the increase in efficiency is 2 per cent for "correct mixture" and 1.5 per cent for "weak mixture."

(4) Fuel consumption (per indicated horse-power per hour) falls, with increased proportions of ether, more rapidly than would be expected from a consideration of the calorific values alone. Between the limits of alcohol alone and alcohol with 15 per cent ether, the fall in consumption (by volume) is as follows:—

4.8	per cent	with	mixture	25 per cent rich
3.8	,,	,,	,,	"correct"
3.3	,,	,,	,,	"weak"

(5) So far as could be observed, the presence of these small percentages of ether appears slightly to have extended the useful range of mixture strength on the weak side, as is evidenced by the increasing length of the power/consumption curves, when the percentage of ether is increased progressively (see Figs. 70–74).

Series III.

Power and Consumption Tests with Varying Load, also with Fixed and Varied Ignition Timing.

The experiments included in this series consisted of tests on the same mixtures of 95 Vol. per cent alcohol and ether, but under three different conditions, namely :—

(A) With full open throttle and constant heat input as for Series II., but with fixed ignition.

(B) At 40 per cent full load torque (brake), 40 per cent of full heat input, and with ignition adjusted.

(C) At 40 per cent full load torque and heating to intake air as for condition (B), but with fixed ignition.

The primary object of these tests was to determine whether the addition of ether would increase the range of burning at reduced loads or would so speed up the rate of burning as to show, with a fixed ignition, a higher thermal efficiency than was obtained with alcohol alone.

These experiments are recorded in detail in the test sheets, pp. 275–286, and on the graphs, Figs. 75 to 86.

Conclusions Based on Tests, Series III.: Under the Full-load Conditions of Tests.

(1) By the additon of ether, the efficiency is increased with fixed ignition as it was with the ignition adjusted. Between the limits of alcohol alone and alcohol with 15 per cent ether, the efficiency, with "weak mixture" and fixed ignition, increased by 2.3 per cent.

(2) The minimum fuel consumption (by volume) decreased by 4.2 per cent as between alcohol alone and a mixture containing 15 per cent of ether when the ignition was fixed.

Under the 40 per cent Load Conditions.

(3) The increase in efficiency due to the addition of ether was, with adjusted ignition, somewhat greater than under the similar full-load conditions.

(4) With fixed ignition the improvement, due to the addition of ether, was slightly less than under the corresponding full-load conditions.

(5) So far as could be ascertained, the presence of ether had but little influence beyond that due to its physical properties as regards heat value and latent heat of evaporation. There was little evidence that at reduced loads when the proportion of residual exhaust products is larger, the ether compensates

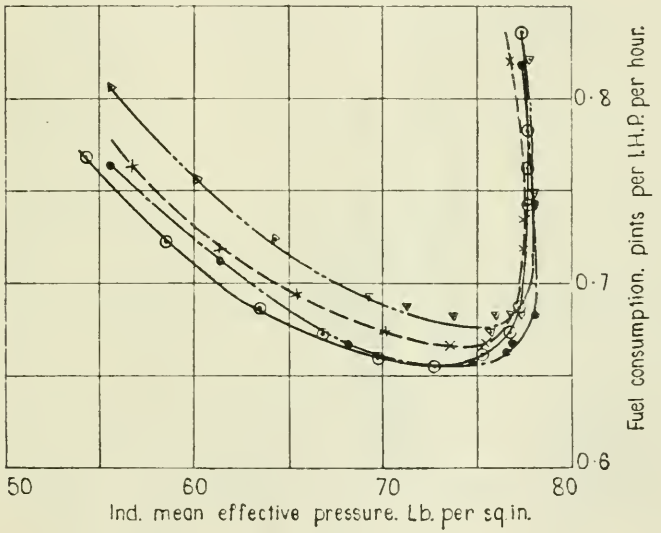
for the reduced rate of burning to an appreciable extent, and thereby renders less important the varying of ignition timing to suit variations of load.

Section II.

The experiments recorded under this section were carried out with a four-cylinder Leyland lorry engine, having a power output of 30 B.H.P. at 1000 revs. per minute.

The bore of the engine is 4.5 in. and the stroke 5 in., or 1320 c.c. per cylinder.

The clearance space in each cylinder was measured, and the values



Series III.—Variable-compression Engine.

Figs. 83, 84, 85 and 86.—Effect on I.M.E.P. and consumption of adding ether to alcohol.

Compression-ratio 5.45 : 1 ; heating to air intake 26 B.Th.Us. per min.; load 40 per cent ; 1500 revs. per minute ; fixed ignition.

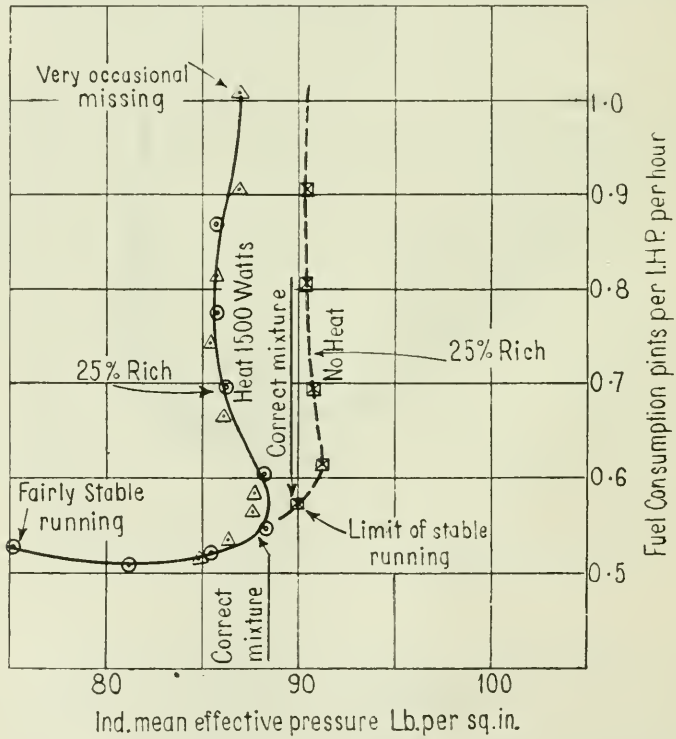
——·——·—— Alcohol 95 Vol. per cent ; jacket temperature 57° C. Test 2.
———————— Alcohol and ether 5 per cent ; jacket temperature 56° C. Test 5.
——·——·——·—— Alcohol and ether 10 per cent ; jacket temperature 57° C. Test 8.
———————— Alcohol and ether 15 per cent ; jacket temperature 56° C. Test 11.

obtained, together with the compression-ratios, are shown in Table XIX.

TABLE XIX.—LEYLAND LORRY ENGINE.

Cylinder.	Clearance vol. c.c.	Compression-ratio
No. 1	417	4.16
No. 2	400	4.30
No. 3	398	4.30
No. 4	411	4.21
	Average compression-ratio	4.24 : 1

The engine was provided with magneto ignition, having arrangements for variable timing, and the carburettor was of the Claudel-



SERIES IV.—LEYLAND LORRY ENGINE.

FIG. 87.—Effect on consumption of variation in mixture strength and heating to air intake with Petrol.

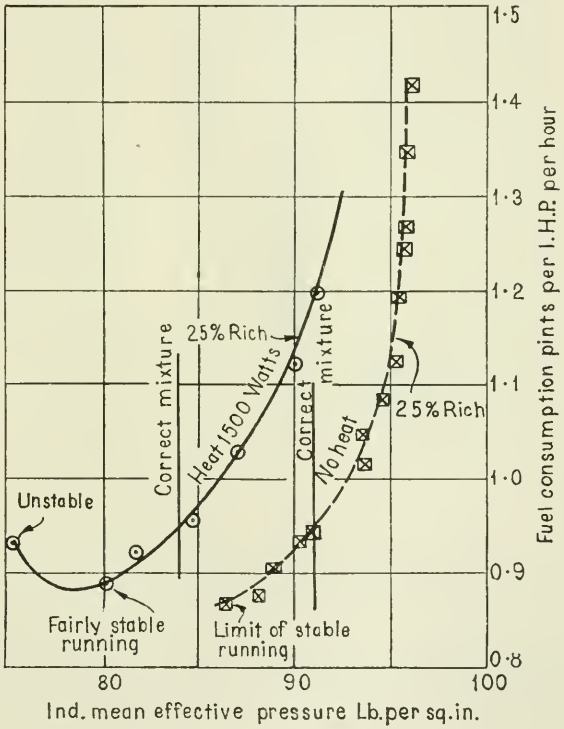
Full load 1000 revs. per minute; heating to air intake 1500 watts; fixed ignition 45 deg. early; water outlet temperature 65° C.

Petrol, A.
Tests 1 and 1A.

Hobson type. An electric heater to the air intake was fitted, and the heat-input could be varied and accurately measured between 2000 watts (114 B.Th.Us. per minute) and zero.

The engine was coupled through a flexible coupling to a balanced-field electric dynamometer, and all arrangements for varying and measuring the load, also for varying and measuring the fuel supplied to the jet of the carburettor, were exactly similar to those used with the single-cylinder variable-compression engine, as described earlier.

Speed.—A number of preliminary tests were made at an engine speed of 1200 revs. per minute, but it was found that with full-



SERIES IV.—LEYLAND LORRY ENGINE.

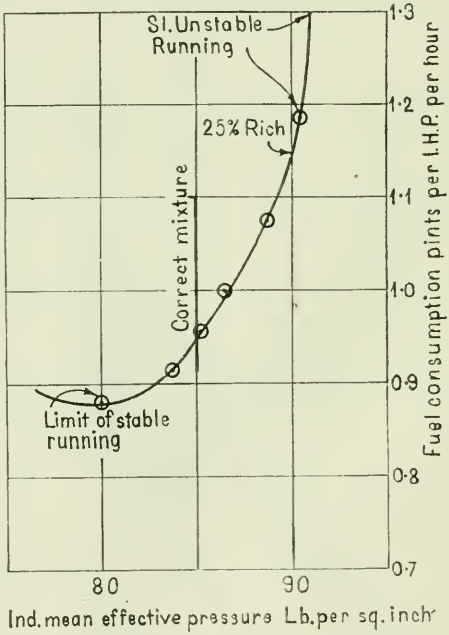
FIG. 88.—Effect on consumption of variation in mixture strength and heating to air intake with Alcohol 95 Vol. per cent.

Full load, 1000 revs. per minute; heating to air intake 1500 watts; fixed ignition 45 deg. early; water outlet temperature 65° C.
Tests 2 and 2A.

open throttle the performance of the engine was not consistent, owing partly to the vibration at this speed, and also probably to the conditions as regards heat-flow to the pistons being too severe for the continuous running required for making power and consumption tests over the whole mixture range for each fuel.

At 1000 revs. per minute the engine was found to behave very well under all conditions of load, and this speed therefore was adopted for the tests which are now described.

Heating.—Preliminary tests were made (these are not here recorded in detail) with alcohol alone and with varying loads to determine the



Series IV.—Leyland Lorry Engine.

Fig. 89.—Effect on consumption of variation in mixture strength with alcohol 95 Vol. per cent and ether 5 Vol. per cent.

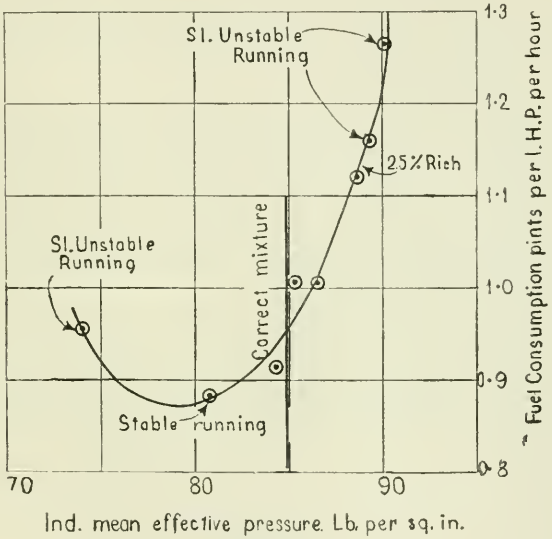
Full load, 1000 revs. per minute; heating to air intake 1500 watts; fixed ignition 45 deg. early; water outlet temperature 65° C. Test 3.

minimum heat which, when applied to the air intake, would give consistent running and an appreciable range of mixture strength.

From a consideration of the practical means employed for carburettor-heating to a lorry or bus engine in service, it was decided

to make the heating during these tests proportional to the square root of the load on the engine, and to find what minimum degree of heating was required when this relation of heating and load was followed.

The preliminary tests showed that with any mixture of alcohol and ether (including alcohol alone), consistent and fairly economical running could be obtained with no heating at full load. At half-load and quarter-load, however, it was found to be impossible, with



Series IV.—Leyland Lorry Engine.

Fig. 90.—Effect on consumption of variation in mixture strength with alcohol 95 Vol. per cent and ether 10 per cent.

Full load, 1000 revs. per minute; heating to air intake 1500 watts; fixed ignition 45 deg. early; water outlet temperature 64° C.
Test 4.

any mixture up to that containing 15 per cent of ether in alcohol, to obtain satisfactory running of the engine. Neither could petrol be used at these reduced loads without heat.

As the result of numerous tests, it was found that the limit appeared at half-load with alcohol alone, and the minimum heat input was discovered in the near neighbourhood of 1000 watts (57 B.Th.Us. per minute) for this load.

Minimum standard heating was therefore fixed upon as in Table XX., the variation, as stated above, being taken as the square root of the brake load :—

TABLE XX.—LEYLAND LORRY ENGINE.

Load		Minimum standard heating for alcohol
Full load (brake)	1500 watts	= 85 B.Th.Us. per min.
Half-load (brake)	1060 ,,	= 60 ,, ,,
Quarter-load (brake)	750 ,,	= 42.5 ,, ,,

In the tests with the four-cylinder engine a procedure was adopted which was entirely similar to that employed in the tests described with the single-cylinder engine.

For Series IV. the throttle was fixed at wide open, while the ignition timing was maintained at 45 degrees before top dead-centre, this angle being measured on the flywheel. The electrical input to the air heater was adjusted for constant heating, and while the mixture strength was varied (by means of a needle valve in the carburettor jet), causing changes of engine output, the field strength of the dynamometer was adjusted, so as to provide a load which maintained constant engine speed.

For Series V. the procedure was similar, except that the throttle was fixed at a point which, for the first five tests, gave half, and for the second five tests gave one-quarter of the full brake horse-power of the engine. Throughout all the tests with the four-cylinder engine, the ignition point was fixed at 45 degrees early, as measured on the crank-angle.

PROGRAMME OF TESTS ON FOUR-CYLINDER (LEYLAND LORRY) ENGINE.

The programme consisted of the following series of tests :—

Series IV.—Consumption and power tests over mixture range at full throttle, with 95 per cent alcohol alone and with 5, 10 and 15 per cent ether : no additional heating to carburettor unless found essential (five complete tests, including a check test, to be run on petrol).

Series V.—Tests for consumption over complete mixture strength at 0.25 and 0.5 of full load, keeping cylinder-jacket temperature constant : no additional heating to carburettor unless found necessary by preliminary tests ; ignition setting to be constant, as found most suitable for full load. Also run one similar check test on petrol at each load (ten complete tests).

Observe and report upon behaviour of four-cylinder engine generally on each mixture, with particular reference to ease of starting, acceleration and " responsiveness " of engine generally.

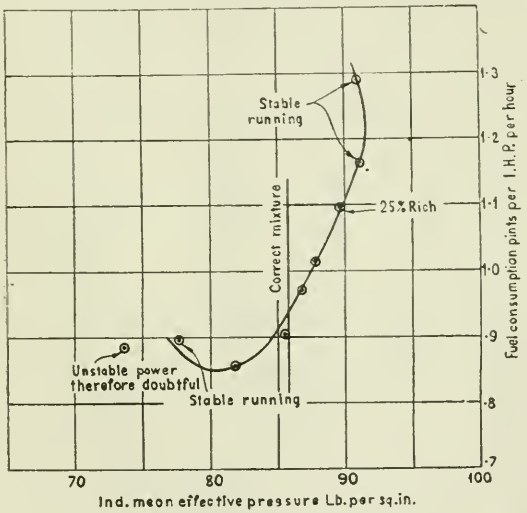
Series IV.

Power and Consumption Tests at Full Load with Leyland Lorry Engine.

The results of these tests are recorded in detail in the test sheets, pp. 287–291, and are shown clearly by the graphs, Figs. 87 to 91.

The conclusions to be drawn may be summarised as follows :—

(1) Under equal conditions of heating, the maximum power falls slightly with increasing proportions of ether. Between the limits of alcohol alone and alcohol with 15 per cent of ether,



Series IV.—Leyland Lorry Engine.

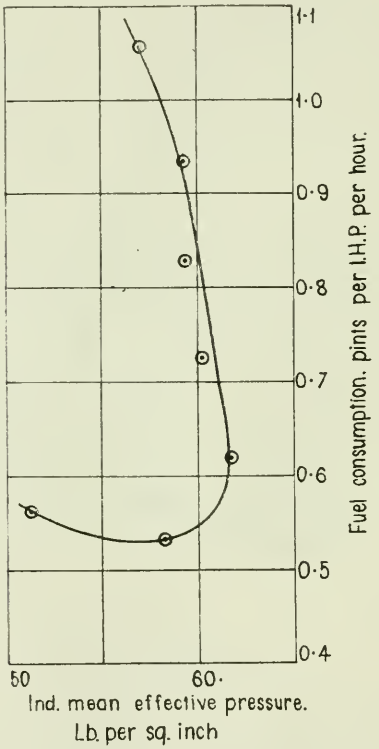
Fig. 91.—Effect on consumption of variation in mixture strength with alcohol 95 Vol. per cent and ether 15 per cent.

Full load, 1000 revs. per minute ; heating to air intake 1500 watts ; fixed ignition 45 deg. early ; water outlet temperature 64° C. Test 5.

the fall in power (on a mixture strength 25 per cent rich) is 1 per cent. A similar result was shown to occur with the single-cylinder engine.

(2) With mixtures 25 per cent rich in each case, the power on alcohol is 5 per cent greater than that obtained with petrol, both for the heated and for the unheated condition. It was shown that with the single-cylinder engine the advantage in power was 8 per cent.

(3) From graph, Fig. 87 (the results being confirmed by three tests made on different days), the maximum power output from petrol was obtained with a mixture only 5 per cent rich and, comparing this power with that obtainable with alcohol



Series V.—Leyland Lorry Engine.

Fig. 92.—Effect on consumption at half load of variation in mixture strength with petrol "A."

Half load, 1000 revs. per minute; heating to air intake 1060 watts; fixed ignition 45 deg. early; water outlet temperature 62° C.
Test 1.

(mixture strength limited to 25 per cent rich), the advantage in favour of alcohol is little more than 2 per cent.

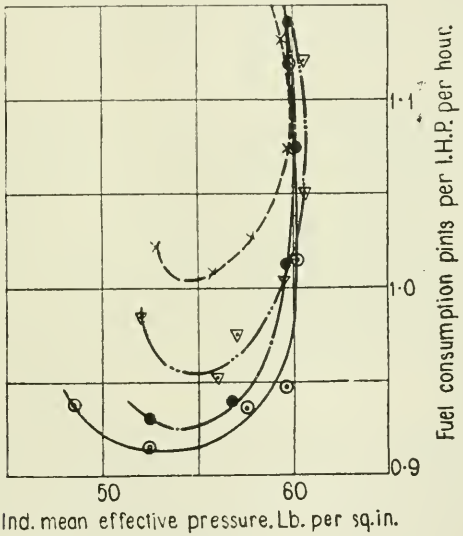
(4) Under full-load conditions there is no measurable increase in efficiency due to the increased proportions of ether in the mixtures with alcohol. This result is quite contrary to expecta-

tion, but it is entirely confirmed both at the "weak" (or most economical) and at the "correct" mixture.

(5) Even the removal of the heating to the air intake caused no measurable fall in efficiency, either at "weak" or at "correct" mixture.

(6) The efficiency on petrol, however, was 7 per cent higher than on alcohol or on the alcohol-ether mixtures. In view of conclusions (4) and (5), this result again is surprising, though it is well confirmed by similar observations made on the tests at half-load.

(7) Fuel consumption under full-load conditions is inversely proportional to the heat value of the alcohol-ether mixtures.



SERIES V.—LEYLAND LORRY ENGINE.

FIGS. 93, 94, 95 AND 96.—Effect on consumption at half load of variation in mixture strength.

Half load, 1000 revs. per minute; heating to air intake 1060 watts; fixed ignition 45 deg. early.

———————— Alcohol 95 Vol. per cent; water outlet temperature 59° C. Test 2.

— - — - — - — Alcohol 95 Vol. per cent. and ether 5 per cent; water outlet temperature 62° C. Test 3.

— - — - — - — Alcohol 95 Vol. per cent. and ether 10 per cent; water outlet temperature 63° C. Test 4.

———————— Alcohol and ether 15 per cent; water outlet temperature 64° C. Test 5.

(8) The consumption on these mixtures as compared with petrol is appreciably less favourable than a mere examination of the relative heat values would indicate.

Series V.

Power and Consumption Tests at Partial Loads with Leyland Lorry Engine (Table XXI.).

From these tests, recorded in detail in the test sheets, pp. 292–301, and on the graphs, Figs. 92 to 101, the following conclusions are drawn :—

(1) Both at half-load and at quarter-load there is a definite advantage as regards power, economy and range of combustion due to the addition of small proportions of ether in alcohol mixtures.

(2) These advantages become well marked for the 10 and 15 per cent ether mixtures. The increase in range of combustion, giving increased "responsiveness" in acceleration, is clearly shown by the graphs of the tests made on half-load—see Figs. 93 to 96. The length of each curve represents the range of mixture over which the engine would run steadily and without "missing."

(3) Between alcohol alone and a mixture containing 10 per cent ether the improvement in efficiency was measured as 7.5 per cent both for half-load and for quarter-load conditions.

(4) At half-load the efficiency on petrol was no less than 12 per cent higher than for the most economical of the alcohol-ether mixtures.

At quarter-load, in the case of the test on the 10 per cent ether mixture, an efficiency superior by 4 per cent (as compared with petrol) was recorded. However, the results obtained at this very small load are given with reservation, since all errors of observation are proportionally magnified and the irregularities of engine running are greatly accentuated as the load is reduced.

General Remarks on the Four-Cylinder Engine Tests.

The results recorded in the preceding paragraphs were considered to be of such great interest as to warrant a fuller investigation. For this purpose we carried out additional tests on the Leyland lorry engine with the air intake connected to the air-measuring apparatus previously described.

Since these further tests were not specifically asked for in connection with the present programme, we have not considered it necessary to include here the detailed results.

Table XXI.—Results with Leyland Lorry Engine.

Speed 1000 *r.p.m., Compression-Ratio* 4.24 : 1.

Test and Series No.	Fuel	Load throttle opening	Heat to air intake B.Th.Us. per min.	Power I.M.E.P. (lb. per sq. in.)		Fuel consumption (pts. per I.H.P. per hr.)			Ind. th. efficiency (per cent.)	
				25 per cent rich	Weak (max. economy)	25 per cent rich	Correct mixture	Weak (max. economy)	Correct mixture	Weak (max. economy)
1 Ser. IV.	Petrol "A"	Full	85	86.0 (max. 88.5)	80.0	0.71	0.55	0.51	25.5	27.5
1A ,, ,,	,, ...	,,	nil	90.5 (max. 91.5)	90.0	0.73	0.56	unstable	25.1	unstable
2 ,, ,,	Alcohol 95 vol. per cent...	,,	85	90.5	79.0	1.15	0.95	0.88	23.8	25.7
2A ,, ,,	,, ,, ,, ..	,,	nil	95.0	86.0	1.15	0.95	0.88	23.8	25.7
3 ,, ,,	Alcoh. ether 5 per cent. ..	,,	85	90.0	79.0	1.14	0.95	0.88	23.7	25.6
4 ,, ,,	,, ,, 10 per cent.	,,	,,	89.0	79.0	1.14	0.95	0.87	23.6	25.8
5 ,, ,,	,, ,, 15 per cent.	,,	,,	89.5	80.0	1.10	0.94	0.85	23.7	26.2
1 Ser. V. ..	Petrol "A"	Half	60	61.0 (max. 61.5)	56.0		—	0.515	—	27.2
2 ,, ,, ..	Alcohol 95 vol. per cent.	,,	,,	59.5	54.0			1.005		22.3
3 ,, ,, ..	Alcoh. ether 5 per cent. ..	,,	,,	60.5	54.5			0.955		23.3
4 ,, ,, ..	,, ,, 10 per cent.	,,	,,	60.0	54.0			0.925		24.0
5 ,, ,, ..	,, ,, 15 per cent.	,,	,,	60.0	53.0			0.915		24.1
6 Ser. V. ..	Petrol "A"	Quarter	42.5	38.5	35.5			0.61		22.9
7 ,, ,, ..	Alcohol 95 vol. per cent.	,,	,,	40.5	37.0			1.01		22.2
8 ,, ,, ..	Alcoh. ether 5 per cent. ..	,,	,,	41.0	36.0			1.04		21.4
9 ,, ,, ..	,, ,, 10 per cent.	,,	,,	42.0	37.0			0.93		23.9
10 ,, ,, ..	,, ,, 15 per cent.	,,	,,	41.5	38.0			0.96		23.0

The conclusions which we are able to draw, however, have a significance in connection with the tests described in this report, and some of the air-measurement results are therefore given in Table XXII.

TABLE XXII.—LEYLAND LORRY ENGINE.

Speed, 1000 revs. per minute, full throttle ; Compression-ratio, 4.24 : 1.

Fuel	Heat to air intake. B.Th.Us. per minute	Mean volumetric efficiency (air at n.t.p.). per cent.	Air consumption (mixture 25 per cent rich)	
			lb. per hour	lb. per I.H.P. per hr.
Petrol "A"	85	62.2	276	8.0 (min. 7.75)
,, ...	nil	65.4	290	7.9 (min. 7.80)
Alcohol 95 Vol. per cent. ..	85	63.5	282	7.55
,, ,, ,, ..	nil	66.5	295	7.65

From these further tests, light is thrown on the reasons for the following facts which were observed, and which have been recorded in this report.

Increase of Maximum Power with Alcohol Mixtures, as compared with Petrol.

It has been recorded above that this increase for the single-cylinder engine was 8 per cent, whereas for the four-cylinder engine it was only 2 per cent.

From Table XXII. it will be seen that the increase in volumetric efficiency, due to the use of alcohol, was only 2 per cent for the lorry engine, while, as we have previously shown, the corresponding increase with the single-cylinder variable-compression engine was very nearly 8 per cent.

From these results arise the general conclusions :—

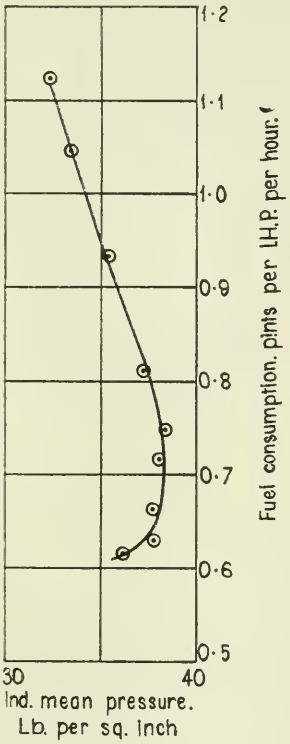
(1) That the extra power on alcohol is due to the corresponding increase in the weight of air drawn into the cylinder.

(2) That this extra weight of air, or increase in volumetric efficiency, varies greatly as between different types of engine.

Loss of Fuel Efficiency (in the Four-Cylinder Engine) with Alcohol Mixtures as compared with Petrol.

It has been recorded (conclusion 6, Series IV, p. 263.) that this loss was measured as 7 per cent.

From Table XXII. it will be seen that the minimum consumption of air per indicated horse-power per hour in the lorry engine was $2\frac{1}{2}$ per cent more favourable with alcohol than with petrol. A similar



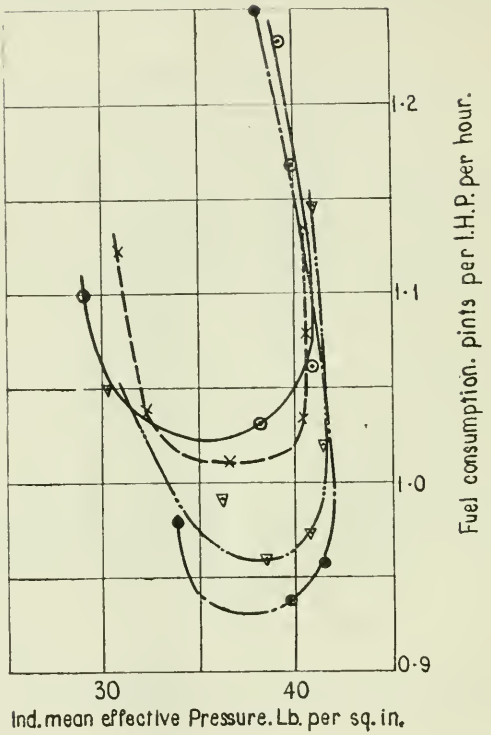
Series V.—Leyland Lorry Engine.

Fig. 97.—Effect on consumption at quarter load of variation in mixture strength with petrol "A."

Quarter load, 1000 revs. per minute; heating to air intake 750 watts; fixed ignition 45 deg. early; water outlet temperature 66° C.
Test 6.

improvement in air efficiency was observed for the single-cylinder variable-compression engine.

Another significant fact, observed with the lorry engine at full load, is that the fuel efficiency with heated air intake was no greater than when running without heat.



SERIES V.—LEYLAND LORRY ENGINE.

FIGS. 98, 99, 100 AND 101.—Effect on consumption at quarter load of variations in mixture strength.

Quarter load, 1000 revs. per minute; heating to air intake 750 watts; fixed ignition 45 deg. early.

———— Alcohol 95 Vol. per cent; water outlet temperature 62° C Test 7.

———— Alcohol 95 Vol. per cent and ether 5 per cent; water outlet temperature 65° C. Test 8.

———— Alcohol 95 Vol. per cent and ether 10 per cent; water outlet temperature 65° C. Test 9.

———— Alcohol 95 Vol. per cent and ether 15 per cent; water outlet temperature 62° C. Test 10.

The following general conclusions therefore are reached :—

(1) That there is an advantage (for both types of engine) in air efficiency by the use of alcohol and its mixtures.

(2) That the rather serious loss in fuel efficiency with the Leyland lorry engine on alcohol or on mixtures containing large proportions of alcohol, is due both to " distribution " losses and to " condensation " in the cylinder. The results of the experiments incline us to the view that condensation—or, more properly, incomplete evaporation or incomplete mixing of the fuel with the air in the cylinder—is mainly responsible for the loss of fuel economy with alcohol.

As concerning the main purpose of this report, the addition of from 10 to 15 per cent of ether in alcohol is justified on the grounds of better performance at reduced loads.

From tests we have made, but which are not recorded here, the temperature at which " starting " is possible will be appreciably lowered by the presence of from 10 to 15 per cent of ether in the alcohol mixture. With such a small proportion of ether, however, it is not possible, with most carburettors, to start the engine from cold.

SERIES II. TEST NO. I.

Consumption and Power at Full Load.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Petrol "A."** *S.G.* 0.785.

Barometer 29.9 in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 38.2 lb.		I.M.E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 65 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
											Volts
1500	3.2	35.0	143.8	34.4	63.0	24.6	16.1	1.2	50.8	—	69*
1478	3.6	34.6	142.4	33.6	59.5	24.5	,,	1.3	43.7	0.614	,,
1492	3.3	34.9	143.4	34.1	60.0	25.0		1.2	45.2	0.585	,,
1496	3.2	35.0	143.8	34.3	61.8	25.3		1.2	48.6	0.540	,,
1533	3.0	35.2	144.5	35.4	64.8	25.5		1.2	50.9	0.500	,,
1536	3.1	35.1	144.1	35.4	66.0	26.6		1.2	53.9	0.472	,,
1532	3.0	35.2	144.5	35.3	65.0	27.0		1.2	52.8	0.483	,,
1536	3.3	34.9	143.4	35.2	67.0	27.0		1.2	55.3	0.462	,,
1478	3.7	34.5	142.0	33.5	61.0	27.5		1.4	59.3	0.453	,,
1480	4.4	33.8	139.5	32.9	61.0	27.3		1.7	60.5	0.452	,,
1482	5.4	32.8	135.9	32.1	59.0	27.6		1.7	64.3	0.436	,,
1498	6.7	31.5	131.3	31.6	58.0	28.1		2.0	67.0	0.425	,,
1485	8.0	30.2	126.6	30.0	—	28.6		2.5	71.0	0.422	,,
1520	10.0	28.2	119.5	29.0	—	28.6		2.8	74.9	0.415	,,
1498	12.7	25.5	109.8	26.3	—	28.7	16.0	3.4	79.5	0.431	,,†

REMARKS.—Friction test, 5.3 lb.; temp., 68° C.

* 21.7 amps. † Very slightly unstable.

SERIES II. TEST No. 2.

Consumption and Power at Full Load.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol 95 Vol. per cent.** *S.G.* 0.8155.

Barometer 29.86in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 50 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input 65 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1502	14.8				56.5	29.2	17.5	1.5	53.0		69*
Alcohol											
1515	11.3	38.7	156.5	38.0	50.0	16.0	,,	1.5	25.5	0.929	,,
1492	11.2	38.8	156.8	37.5	51.0	16.0	,,	1.5	27.0	0.889	,,
1490	11.5	38.5	155.8	37.2	51.8	16.0	,,	,,	29.0	0.835	,,
1505	11.7	38.3	155.1	37.4	53.0	16.2	,,	,,	30.3	0.795	,,
1508	11.9	38.1	154.3	37.3	54.7	16.2	,,	,,	32.8	0.736	,,
1480	12.8	37.2	151.1	35.7	54.9	16.2	,,	1.6	36.0	0.700	,,
1518	14.6	35.4	144.7	35.2	54.9	16.0	,,	2.0	37.8	0.676	,,
1486	15.9	34.1	140.1	33.4	50.6	15.9	,,	2.0	40.2	0.671	,,
1490	16.9	33.1	136.5	32.6	49.5	15.9	,,	2.0	41.4	0.667	,,
1497	18.7	31.3	130.1	31.2	58.0	16.0	,,	2.5	43.6	0.662	,,
1482	21.6	28.4	119.8	28.5	49.0	16.0	,,	3.8	46.2	0.683	67.5†

REMARKS.—Friction test, 5.2 lb.; temp., 62° C.

* 21.7 amps. † 21.3 amps.

SERIES II. TEST NO.

Consumption and Power at Full Load.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 5 per cent.** *S.G.* 0.813.

Barometer 29.86in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 50 lb.		I.M.E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 65 B.Th.Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1500	14.6				56.0	27.0	17.5	1.5	52.0		69*
Alcohol											
1506	11.6	38.4	155.6	37.5	51.0	16.0	17.5	1.5	27.1	0.886	,,
1488	11.8	38.2	154.9	36.8	53.0	15.5	,,	1.5	31.3	0.781	,,
1483	11.8	38.2	154.9	36.7	53.5	15.2	,,	,,	34.0	0.722	,,
1480	12.5	37.5	152.3	36.1	54.0	15.8	,,	,,	36.5	0.683	,,
1510	14.4	35.6	145.5	35.2	52.0	15.5	,,	1.4	38.0	0.673	,,
1476	15.4	34.6	141.9	33.5	50.5	15.6	,,	1.7	40.5	0.664	,,
1482	16.3§	32.9§	138.7	34.2	53.2	15.6	,,	1.8	41.6	0.658	,,
1488	18.0	32.0	132.6	31.6	51.2	15.5	,,	2.3	43.7	0.653	,,
1502	20.5	29.5	123.6	29.8	51.8	15.5	,,	3.0	46.2	0.654	68†
1499	22.5	27.5	116.4	28.0	50.8	,,	17.2	3.8	46.6	0.690	66‡

REMARKS.—Friction 5.2 lb., as per 95 per cent alcohol test.

* 21.7 amps. † 21.5 amps. ‡ 20.2 amps. § Evident error.

SERIES II. TEST NO. 4.

Consumption and Power at Full Load.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 10 per cent.** *S.G.* 0.810.

Barometer 29.86in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 50 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 65 B.Th.Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"*A*"										Volts
1500	14.5				52.5	25.0	17.0	1.5	50.5		69*
Alcohol											
1515	11.5	38.5	155.7	37.8	49.5	14.6	,,	,,	26.9	0.885	,,
1505	11.7	38.3	155.0	37.4	51.5	14.8	,,	,,	30.3	0.794	,,
1466	11.7	38.3	155.0	36.4	52.5	15.0	,,	,,	32.9	0.752	,,
1485	12.3	37.7	152.8	36.4	54.0	15.0	,,	1.8	35.1	0.705	,,
1478	13.3	36.7	149.3	35.4	53.6	15.0	,,	1.8	37.5	0.678	,,
1486	15.1	34.9	142.8	34.8	52.0	15.1	,,	2.1	39.6	0.653	,,
1480	17.2	32.8	135.3	32.1	54.5	15.2	,,	3.0	43.0	0.652	,,
1515	18.7	31.3	129.9	31.6	54.0	15.2	17.3	3.5	44.2	0.645	,,
1502	20.5	29.5	123.4	29.8	52.5	15.2	,,	3.8	47.0	0.643	68†
1502	24.0	26.0	110.8	26.8	50.5	15.2	,,	3.8	47.8	0.703	67‡

REMARKS.—Friction assumed to be 5.2.

* 21.7 amps. † 21.5 amps. ‡ 21.3 amps., pop. back.

SERIES II. TEST NO. 5.

Consumption and Power at Full Load—Ignition Varied.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 15 per cent.** *S.G.* 0.8055.

Barometer 29.85in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 50 lb		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 65 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1500	14.3				54.0	23.6	14.0	1.5	50.5		69*
Alcohol											
1481	11.5	38.5	155.4	37.0	51.9	13.0	,,	,,	29.5	0.825	,,
1516	11.5	38.5	155.4	37.8	51.5	13.2	,,	,,	32.3	0.738	,,
1483	12.0	38.0	153.6	36.6	55.0	13.5	,,	,,	35.8	0.687	,,
1485	12.5	37.5	151.8	36.2	54.0	14.0	,,	,,	37.2	0.669	,,
1494	13.7	36.3	147.5	35.4	53.5	13.8	,,	1.6	38.4	0.662	,,
1486	15.1	34.9	142.5	34.0	52.0	14.0	,,	2.0	40.4	0.654	,,
1504	16.0	34.0	139.3	33.7	51.5	14.0	15.0	2.5	41.0	0.652	,,
1495	18.0	32.0	132.2	31.7	48.5	14.0	,,	3.0	44.3	0.641	,,
1499	20.3	29.7	124.0	29.9	51.0	14.0	,,	,,	47.1	0.639	68†
1500	24.3	25.7	110.0	26.5	50.5	14.0	15.0	3.0	48.0	0.708	67‡

REMARKS.—Friction taken to be 5.2 lb.

* 21.7 amps. † 21.4 amps. ‡ 21 amps.

SERIES III. TEST No. 1.

Consumption and Power—Full Load—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol 95 Vol. per cent.** *S.G.* 0.8155.

Barometer 30.45in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 50 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint.		Heat input 65 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "*A*"											Volts..
1500	14.2				59.0	25.8	15.3	1.5	50.1		69*
Alcohol											
1504	11.1	38.9	154.5	38.0	56.5	16.5	,,	1.5	32.1	0.740	69*
1504	10.8	39.2	155.5	38.3	56.8	16.5	,,	1.5	30.8	0.764	
1476	11.1	38.9	154.5	37.3	57.5	16.3	,,	,,	33.0	0.732	
1501	11.6	38.4	152.8	37.6	,,	,,	,,	,,	34.4	0.696	68†
1484	12.8	37.2	148.6	36.1	,,	,,		,,	36.4	0.685	
1480	14.0	36.0	144.3	34.9	55.0	16.8		,,	37.8	0.682	
1504	15.5	34.5	139.1	34.2	58.5	,,		,,	39.1	0.674	
1521	16.3	33.7	136.2	34.0	57.8	,,		,,	39.6	0.669	
1501	18.2	31.8	128.6	31.9	61.2	16.5		,,	41.9	0.675	
1496	20.2	29.8	122.6	29.6	53.0	16.3	16.3	,,	43.5	0.700	
1512	22.0	28.0	116.3	28.8	,,	,,	,,	,,	44.0	0.710	‡

REMARKS.—Friction test, 5.4 lb.; temp., 58° C.; had to motor for a long time to get to speed, owing to insufficient power.

* 21.7 amps. † 21.5 amps. ‡ Slightly unstable.

SERIES III. TEST NO. 4.

Consumption and Power—Full Load—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 5 per cent.** *S.G.* 0.813.

Barometer 30.46in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft rad. W = 50.2 lb. Spring balance	Nett weight (= w)	I.M. E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint Seconds unit vol	Pint per I.H.P. per hr.	Heat input 65 B.Th. Us. per min. (net)
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1500	14.5				58.0	29.5	23.0	1.5	49.5		69*
Alcohol											
1471	11.2	39.0	154.5	37.3	54.0	18.2	23.0	1.5	31.8	0.760	
1475	11.6	38.6	153.3	37.0	56.0	18.0		1.5	34.5	0.705	
1491	13.2	37.0	147.6	36.1	56.0	18.2		,,	36.7	0.680	
1480	15.2	35.0	141.0	34.2	54.3	18.1		,,	39.4	0.668	
1480	18.2	32.0	130.3	31.6	53.5	18.2	24.0	,,	42.5	0.670	
1496	20.6	29.6	122.0	29.9	53.5	18.3		,,	44.6	0.675	
1483	22.7	27.5	114.6	27.9	53.0	18.2		,,	45.5	0.709	67

REMARKS.—Friction test, 5.4 lb.; temp., 68° C.

* 21.6 amps.

SERIES III. TEST No. 7.

Consumption and Power—Full Load—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 10 per cent.** *S.G.* 0.810.

Barometer 30.19in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 50 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 65 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1500	14.5				65.0	27.0	17.0	1.5	49.8		69*
Alcohol											
1515	12.2	37.8	152.0	37.3	63.0	15.5	17.0	1.5	35.0	0.690	69*
1513	14.5	35.5	144.0	35.3	56.0	15.5	,,	1.5	38.4	0.665	
1521	15.8	34.2	139.5	34.1	52.0	15.5	,,	,,	40.2	0.657	
1502	16.5	33.5	137.0	33.3	49.0	15.8	,,	,,	41.4	0.654	
1528	20.4	29.6	123.0	30.5	52.0	15.5	,,	,,	44.3	0.657	
1503	23.3	26.7	112.5	27.5	51.0	15.5	,,	,,	46.3	0.707	66†

REMARKS.—Dynamometer balance checked. Friction test, 5.3 lb.; temp., 65° C

* 21.7 amps. † 20.7 amps.

SERIES III. TEST No. 10.

Consumption and Power—Full Load—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 15 per cent.** *S.G.* 0.8055.

Barometer 30.16in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 50 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 65 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "*A*"											Volts
1500	14.2				50.0	28.0	20.0	1.5	51.9		69*
Alcohol											
1492	11.8	38.2	153.8	37.1	52.0	16.0	,,	,,	32.5	0.746	69*
1515	12.3	37.7	152.0	37.2	52.5	16.0	20.5	,,	35.5	0.681	,,
1505	13.2	36.8	148.5	36.2	52.0	16.0	,,	,,	37.2	0.668	,,
1499	14.6	35.4	143.8	34.8	49.5	16.3	,,	,,	39.4	0.656	,,
1482	16.1	33.9	138.4	33.2	50.5	16.5	,,	,,	42.3	0.641	,,
1487	18.5	31.5	130.0	31.3	51.8	16.5	20.1	,,	44.4	0.647	,,
1491	22.2	27.8	117.0	28.2	51.8	16.5	,,	,,	47.2	0.676	,,

REMARKS.—Friction test, 5.3 lb.; temp., 60° C. Dynamometer balance checked.

* 21.7 amps.

SERIES III. TEST NO. 3.

Consumption and Power—Load 40 per cent.—Ignition Adjusted.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol 95 Vol. per cent.** *S.G.* 0.8155.

Barometer 30.52in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20 lb.		I.M.E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption Unit vol. ¼ pint		Heat input 26 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1500	5.3	14.7	73.5	17.9	55.0	38.0	22.3	2.0	111.5	0.451	42*
Alcohol											
1492	4.3	15.7	76.7	18.7	53.6	18.5	,,	2.0	65.5	0.735	,,
1492	4.1	15.9	77.4	18.8	55.0	18.0	,,	2.0	58.4	0.820	,,
1493	4.1	15.9	77.4	18.8	56.0	18.0	,,	2.0	61.6	0.778	,,
1503	4.2	15.8	77.1	18.9	57.8	18.0	,,	2.0	66.6	0.716	,,
1487	4.7	15.3	75.3	18.3	57.4	18.0	,,	2.4	72.2	0.682	,,
1489	5.2	14.8	73.6	17.9	57.0	18.0	24.3	2.7	74.5	0.675	,,
1482	5.5	14.5	72.5	17.5	56.0	18.0	,,	2.9	76.5	0.672	,,
1482	6.1	13.9	70.4	17.0	56.5	18.0	,,	3.2	79.1	0.670	,,
1488	7.3	12.7	66.4	16.1	57.5	17.9	,,	3.8	81.7	0.684	,,

REMARKS.—Friction test, 6.2 lb.; temp., 68° C.

* 14.2 amps.

SERIES III. TEST NO. 6.

Consumption and Power—Load 40 per cent.—Ignition Adjusted.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 5 per cent.** *S.G.* 0.813.

Barometer 30.44in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20.2 lb. Spring balance	Nett weight (= w)	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint Seconds unit vol	Pint per I.H.P. per hr.	Heat input 26 B.Th. Us. per min. (net)
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1500	5.7				49.0	37.6	21.0	2.0	112.2		42*
Alcohol											
1507	4.3	15.9	76.9	18.8	52.0	16.9	,,	2.2	55.6	0.861	
1527	4.3	15.9	76.9	18.8	53.5	17.0	,,	,,	57.8	0.829	
1517	4.4	15.8	76.5	18.9	54.0	17.0	,,	,,	58.4	0.816	
1531	4.4	15.8	76.5	18.9	56.0	17.5		,,	63.5	0.750	
1525	4.5	15.7	76.2	18.8	57.0	17.6		2.3	69.9	0.685	
1522	4.8	15.4	75.1	18.6	57.5	17.6		2.5	71.4	0.678	
1530	5.1	15.1	74.1	18.4	57.6	17.6		2.6	72.5	0.675	
1515	5.5	14.7	72.7	17.9	56.0	17.5		3.0	75.4	0.667	
1522	6.1	14.1	71.6	17.5	55.0	17.6		3.5	77.3	0.665	
1515	7.0	13.2	67.4	16.6	54.0	17.5		3.8	80.2	0.676	
1529	7.2	13.0	66.7	16.5	54.0	17.3		3.8	80.4	0.679	†

REMARKS.—Friction taken as 6.0 lb.

* 14.2 amps. † Slightly unstable.

SERIES III. TEST No. 9.

Consumption and Power—Load 40 per cent.—Ignition Adjusted.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 10 per cent.** *S.G.* 0.810.

Barometer 30.2in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20 lb.		I.M. E.P.	Ind. H.P.	Temperature °C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 26 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1500	5.0	15.0	74.0		61.4	37.9	18.0	2.5	113.7		42*
Alcohol											
1531	4.3	15.7	76.6	19.0	51.5	16.0	,,	2.4	57.7	0.821	
1514	4.2	15.8	77.0	18.5	53.7	16.1	,,	2.4	62.8	0.775	
1520	4.1	15.9	77.3	19.0	54.8	16.5	,,	,,	68.8	0.689	
1516	4.2	15.8	77.0	18.9	55.0	16.7	,,	,,	70.3	0.677	
1506	4.5	15.5	75.9	18.5	55.2	16.6	,,	2.6	72.8	0.668	
1520	5.1	14.9	73.8	18.1	55.2	16.5	,,	2.8	76.7	0.649	
1515	5.8	14.2	71.4	17.5	55.0	16.5	,,	3.5	79.6	0.646	
1512	6.8	13.2	67.8	16.6	55.0	16.3	20.1	3.8	82.2	0.660	
1513	8.5	11.5	61.8	15.1	49.0	16.8	,,	3.8	85.8	0.694	

REMARKS.—Dynamometer balance checked. Friction assumed at 6.0 lb. = 21.3 lb. M.E.P.

* 14.2 amps.

SERIES III. TEST NO. 12.

Consumption and Power†—Load 40 per cent.—Ignition Varied.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 15 per cent.** *S.G.* 0.8055.

Barometer 29.97in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20 lb.		I.M.E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 26 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1500	5.0				58.9	37.6	18.6	2.0	108.8		42*
Alcohol											
1528	4.4	15.6	77.1	18.9	55.3	15.2	,,	2.0	62.2	0.765	
1494	4.4	15.6	77.1	18.5	55.0	15.0	,,	2.0	58.5	0.819	
1531	4.3	15.7	77.4	19.1	55.5	15.2	,,	2.0	66.6	0.706	
1528	4.3	15.7	77.4	19.0	56.0	15.2		2.0	65.8	0.719	
1518	4.3	15.7	77.4	18.9	57.0	15.5		2.0	71.8	0.664	
1516	4.5	15.5	76.7	18.7	57.0	15.5		2.2	74.3	0.649	
1520	4.7	15.3	76.0	18.6	57.5	15.5		2.5	75.0	0.646	
1511	6.3	13.7	70.4	17.1	53.5	15.2		3.0	83.6	0.630	
1506	6.7	13.3	68.9	16.7	53.0	15.3		3.0	84.0	0.645	
1504	6.0	14.0	71.8	17.3	54.0	15.5		3.8	82.2	0.633	
1496	7.1	12.9	67.5	16.2	53.0	16.0		3.8	86.3	0.644	

REMARKS.—† 40 per cent heat. Dynamometer balance checked. Friction taken as 6.1 lb.

* 14.2 amps.

SERIES III. TEST NO. 2.

Consumption and Power—Load 40 per cent.—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol, 95 Vol. per cent.** *S.G.* 0.8155.

Barometer 30.53in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20.2 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 26 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1523	4.8	15.4	75.7	18.6	58.0	36.5	18.3	2.0	109.6	0.442	42*
Alcohol											
1504	4.8	15.4	75.7	18.4	68.0	17.3	,,	2.0	72.5	0.675	
1501	5.4	14.8	73.6	17.8	55.0	17.0	,,	2.0	73.9	0.684	
1528	4.3	15.9	77.4	19.1	58.2	17.0	,,	,,	68.5	0.688	
1523	4.3	15.9	77.4	19.1	59.3	17.2	,,	,,	68.9	0.684	
1518	4.4	15.8	77.1	18.9	59.3	17.3	,,	,,	69.6	0.684	
1526	4.7	15.5	76.0	18.7	59.5	17.5	,,	,,	70.5	0.683	
1529	5.5	14.7	73.2	18.1	58.3	17.2	19.0	,,	73.3	0.678	
1516	6.1	14.1	71.2	17.4	57.0	17.2	,,	,,	75.2	0.688	
1524	6.7	13.5	69.1	17.0	56.5	17.2	,,	,,	76.4	0.693	
1495	8.1	12.1	64.2	15.5	57.3	17.2	,,	,,	80.3	0.723	
1505	9.3	10.9	60.0	14.5	56.8	17.3	,,	,,	82.3	0.754	
1519	10.6	9.6	55.5	13.5	56.8	17.4	,,	,,	82.8	0.805	
1505	4.1	16.1	78.0	19.1	56.0	17.3	,,	,,	62.8	0.750	
1502	4.2	16.0	77.6	19.0	57.1	17.3	,,	,,	57.8	0.821	

REMARKS.—Throttle closed slightly after taking readings on petrol "A" in order to bring power to 40 per cent of brake load at full throttle. Friction test (throttled), 6.2 lb.; temp. = 69° C.

* 14.2 amps.

SERIES III. TEST NO. 5.

Consumption and Power—Load 40 per cent.—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 5 per cent.** *S.G.* 0.813.

Barometer 30.44in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20.2 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ½ pint		Heat input 26 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1505	5.3				56.5	41.8	21.9	2.0	103.5		42*
Alcohol											
1525	4.3	15.9	77.4	19.3	63.0	17.0	21.8	2.0	63.5	0.735	
1511	4.5	15.7	76.7	19.0	59.0	17.0	,,	,,	58.1	0.820	
1525	4.3	15.9	77.4	19.2	60.5	17.6	,,	,,	65.2	0.719	
1512	4.9	15.3	75.3	18.6	57.5	17.7	,,	,,	72.4	0.669	
1511	5.4	14.8	73.5	18.2	56.0	17.6	,,	,,	74.3	0.666	
1515	6.3	13.9	70.4	17.4	55.0	17.5	,,	,,	76.9	0.675	
1520	7.7	12.5	65.5	16.2	53.5	17.5	,,	,,	80.2	0.695	
1506	8.9	11.3	61.3	15.1	58.0	17.5	,,	,,	83.5	0.717	
1510	10.2	10.0	56.7	14.0	58.3	17.3	,,	,,	84.9	0.761	†

REMARKS.—Friction taken as 6.2 lb.

* 14.2 amps. † Slightly unstable. Popping back when weakened.

SERIES III. TEST No. 8.

Consumption and Power—Load 40 per cent.—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 10 per cent.** *S.G.* 0.810.

Barometer 30.19in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption.		Heat input 26 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Unit vol. Seconds unit vol	¼ pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1508	4.9				60.2	37.9	17.0	2.0	103.4		42*
Alcohol											
1524	4.2	15.8	77.3	18.9	55.5	15.8	,,	,,	58.3	0.816	
1541	4.0	16.0	78.0	19.3	57.2	15.8	,,	,,	62.8	0.743	
1531	4.0	16.0	78.0	19.1	59.0	15.8	,,	,,	69.0	0.683	
1522	4.3	15.7	76.9	18.8	59.0	16.0	,,	,,	71.9	0.666	
1516	4.4	15.6	76.6	18.6	58.0	16.0	,,	,,	72.8	0.664	
1533	4.8	15.2	75.1	18.5	58.5	16.0	,,	,,	73.8	0.659	
1506	5.7	14.3	72.0	17.7	56.5	16.2	,,	,,	77.9	0.652	
1518	6.8	13.2	68.0	16.6	58.3	16.2	17.9	,,	81.1	0.668	
1515	8.7	11.3	61.3	14.9	59.0	16.2	,,	,,	84.9	0.712	
1510	10.2	9.8	55.5	13.6	58.0	16.1	,,	,,	86.4	0.765	†

REMARKS.—Dynamometer balance checked. Friction test, 5.95 lb.; temp., 68.8° C.

* 14.2 amps. † Unstable. Popped back when weakened.

SERIES III. TEST NO. 11.

Consumption and Power—Load 40 per cent.—Fixed Ignition.

Compression-Ratio 5.45 : 1.

Air Standard Efficiency 49.4 per cent.

Fuel—**Alcohol and Ether, 15 per cent.** *S.G.* 0.8055.

Barometer 30.05in. *Ignition Position* No. 0 = 26 deg. early.

Revs. per min.	Torque at 3ft. rad. W = 20 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input 26 B.Th. Us. per min. (net)
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1508	5.6				47.0	38.5	17.0	1.5	112.4		42*
Alcohol											
1522	4.3	15.7	77.3	18.9	46.0	14.0	,,	,,	57.2	0.833	
1536	4.2	15.8	77.6	19.2	52.5	14.2	,,	,,	61.6	0.761	
1530	4.2	15.8	77.6	19.1	52.6	14.4	,,	,,	60.4	0.781	
1521	4.2	15.8	77.6	19.0	54.8	14.7	,,	,,	62.2	0.763	
1513	4.2	15.8	77.6	18.9	55.0	14.8	,,	,,	64.4	0.741	
1512	4.3	15.7	77.3	18.8	55.8	15.0	,,	,,	69.4	0.689	
1525	4.4	15.6	76.9	18.9	56.8	15.1	,,	,,	70.7	0.674	
1516	4.9	15.1	75.1	18.4	56.5	15.2	,,	,,	74.1	0.661	
1521	5.6	14.4	72.7	17.8	56.0	15.0	,,	,,	77.2	0.656	
1518	6.4	13.6	69.9	17.1	54.7	15.0	,,	,,	79.7	0.660	
1520	7.3	12.7	66.7	16.3	53.5	15.0	,,	,,	82.1	0.673	
1529	8.2	11.8	63.5	15.6	53.0	15.0	18.0	,,	84.2	0.686	
1511	9.6	10.4	58.5	14.2	52.2	15.0	,,	,,	86.7	0.721	†
1520	10.8	9.2	54.3	13.3	51.0	15.0	,,	,,	88.3	0.766	‡

REMARKS.—Dynamometer balance checked. Friction test, 6.1 lb.; temp., 63° C.

* 14.2 amps. † Slightly unstable. ‡ Unstable.

LEYLAND ENGINE.

SERIES IV. TEST No. 1.

Consumption and Power at Full Load.

Fuel—**Petrol "A."** *S.G.* 0.785.

Barometer 29.97in.

NOTE.—Throughout the following tests on the Leyland lorry engine, the expansion-ratio was 4.24 : 1 ; the ignition position was fixed, 45 deg. early ; and the air standard efficiency was taken as 44 per cent.

Revs. per min.	Torque at 3ft. rad. W = 64 lb. Spring balance	Nett weight (= w)	I.M.E.P.. lb. per sq. in.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint Seconds unit vol	Pints per I.H.P. per hr.	Heat input (1500 watts)
1	2	3	4	5	6	7	8	9	10	11	12
											Volts
1000	18.1	45.9	84.4	34.0	65.0		21.3	4.5	31.3	0.846	100*
1010	17.4	46.6	85.4	34.7	65.5		21.3	,,	27.6	0.940	
1000	17.1	46.9	85.8	34.5	67.0		21.3	,,	30.0	0.870	
995	17.1	46.9	85.8	34.4	67.0		21.7	,,	33.8	0.774	
1000	16.8	47.2	86.2	34.7	66.5		21.5	,,	37.5	0.692	
1000	15.5	48.5	88.1	35.5	65.3		21.5	,,	42.0	0.604	
1000	15.4	48.6	88.2	35.5	65.3		22.0	,,	46.2	0.549	
1005	17.4	46.6	85.4	34.5	62.3		21.8	,,	50.0	0.522	
1000	20.3	43.7	81.3	32.7	64.5		22.0	,,	54.0	0.510	
1010	24.6	39.4	75.2	30.6	63.5		22.0	,,	56.0	0.525	†
				motoring hot							
970		13.5									
1030		13.7									

REMARKS.—Water temp., 61/71° C. after motoring. Friction, 13.6 = 19.3 I.M.E.P.

* 15 amps. † Very occasional missing.

LEYLAND ENGINE.

SERIES IV. TEST NO. 2.

Consumption and Power at Full Load.

Fuel—**Alcohol 95 Vol. per cent.** *S.G.* 0.8155.

Barometer 30.07in.

Revs. per min.	Torque at 3ft. rad. W = 64 lb. Spring balance	Torque at 3ft. rad. W = 64 lb. Nett weight (= w)	I.M.E.P., lb. per sq. in.	Ind. H.P.	Temperature ° C. Inlet housing Water outlet	Temperature ° C. Inlet housing Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint Seconds unit vol	Fuel consumption. Unit vol. ¼ pint Pint per I.H.P. per hr.	Heat input (1500 watts)
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1000	16.8	47.2			59.5		24.2		33.7		100*
Alcohol											
1000	12.4	51.6	90.8	36.6	62.0		24.3	4.5	20.5	1.199	†
1000	13.0	51.0	90.0	36.2	62.4		24.5	,,	22.2	1.12	
1010	14.5	49.5	87.8	35.7	62.6		24.8	,,	22.6	1.113	
1000	15.1	48.9	87.0	35.0	63.0		24.8	,,	25.0	1.024	
1000	16.8	47.2	84.6	34.1	62.0		24.8	,,	27.6	0.957	
1000	18.8	45.2	81.8	33.0	64.0		24.6	,,	29.6	0.920	
1000	19.9	44.1	80.2	32.3	64.0		24.7	,,	31.5	0.885	‡
1000	23.5	40.5	75.1	30.3	64.5		24.7	,,	31.9	0.932	§

REMARKS.—Friction, 12.5 lb. = 17.8 F.M.E.P.; water temp., 86.5° C.

* 15 amps. † Occasionally missing. ‡ Slightly unstable. § Very unstable, missing and popping back.

LEYLAND ENGINE.

SERIES IV. TEST NO. 3.

Consumption and Power at Full Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 5 per cent.** *S.G.* 0.813.
Barometer 30.07in.

Revs. per min.	Torque at 3ft. rad. W = 64 lb. Spring balance	Nett weight (= w)	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ¼ pint Seconds unit vol	Pint per I.H.P. per hr.	Heat input, 1500 watts
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "*A*"											Volts
1000	16.5	47.5			62.0		23.8	4.5	32.8		95*
Alcohol Ether Mixture											
1010	12.8	51.2	90.3	36.8	61.5		24.3	,,	20.6	1.187	†
1010	13.2	50.8	89.7	36.5	63.5		24.8	,,	22.0	1.120	
1000	13.9	50.1	88.8	35.7	63.5		25.0		23.4	1.077	100‡
1010	15.4	48.6	86.5	35.3	63.5		25.0		25.5	1.000	
1015	16.3	47.7	85.3	35.0	62.5		25.0		26.7	0.962	
1000	17.5	46.5	83.6	33.7	61.5		25.0		29.3	0.912	
1000	20.0	44.0	80.1	32.3	65.5		25.0		31.7	0.880	†

REMARKS.—Friction, 12.5 lb. = 17.8 M.E.P., temp., 85.5° C.

* 14 amps. † Occasional missing. ‡ 15 amps.

LEYLAND ENGINE.

SERIES IV. TEST NO. 4.

Consumption and Power at Full Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 10 per cent.** *S.G.* 0.810.

Barometer 30.06in.

Revs. per min.	Torque at 3ft. rad. W = 64 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 1500 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
995	17.4	46.6			68.0		25.3	4.5	33.7		100*
Alcohol Ether Mixture											
1000	13.5	50.5	90.0	36.3	59.0		25.2	,,	19.6	1.265	†
1000	13.7	50.3	89.2	36.0	62.6		25.4	,,	21.5	1.161	‡
1000	14.2	49.8	88.5	35.7	64.5		25.6	,,	22.5	1.120	
990	15.7	48.3	86.4	34.5	,,		,,	,,	25.9	1.006	
1010	16.5	47.5	85.2	34.2	,,		,,	,,	26.1	1.008	
1000	17.2	46.8	84.3	34.0	63.5		25.7	,,	29.0	0.913	
995	19.8	44.2	80.6	32.3	61.5		26.0	,,	31.6	0.883	
995	24.5	39.5	74.0	29.6	60.0		26.0	,,	32.9	0.956	§

REMARKS.—Friction test, 12.7 lb. = 18.1 lb. F.M.E.P.

* 13 amps. † Occasional missing. ‡ Very occasional missing. § Slightly unstable and missing.

LEYLAND ENGINE.

SERIES IV. TEST No. 5.

Consumption and Power at Full Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 15 per cent.** *S.G.* 0.8055.

Barometer 30.06in.

Revs. per min.	Torque at 3ft. rad. W = 64 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 1500 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1010	16.4	47.6			64.0		25.2	4.5	31.9		100*
Alcohol Ether Mixture											
1010	12.5	51.5	91.0	37.0	61.5		26.0	,,	18.9	1.290	
1015	12.4	51.6	91.1	37.2	62.5		26.0	,,	20.8	1.161	
1000	13.4	50.6	89.7	36.2	64.0		26.3	,,	22.6	1.099	
1000	14.7	49.3	87.8	35.4	66.0		25.5	,,	25.0	1.016	
1000	15.5	48.5	86.7	35.0	65.5		26.3	,,	26.5	0.971	
1000	16.4	47.6	85.5	34.4	65.0		26.6	,,	29.0	0.902	
1005	19.0	45.0	81.8	33.1	63.0		26.5	,,	31.8	0.855	
1000	21.8	42.2	77.8	31.3	60.0		26.5	,,	32.2	0.893	
995	24.7	39.3	73.6	29.5	62.0		26.5	,,	34.6	0.882	†

REMARKS.—Friction test, 12.7 lb. = 18.1 lb. F.M.E.P.; temp., 76.5° C.

* 15 amps. † Unstable.

LEYLAND ENGINE.

SERIES V. TEST No. 1.

Consumption and Power at Half Load.

Fuel—Petrol "A." *S.G.* 0.735.

Barometer 30.05in.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ‡ pint		Heat input, 1060 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
											Volts
1000	13.1	26.9	57.0	22.6	57.5		21.0		37.6	1.059	84.5*
1000	11.4	28.6	59.4	23.6	55.0		21.0		40.8	0.935	
1000	11.4	28.6	59.4	23.6	55.0		21.4		46.0	0.829	
1000	10.9	29.1	60.2	23.9	61.0		—		51.9	0.725	
1000	9.8	30.2	61.7	24.5	63.7		21.5		59.3	0.620	
1012	12.1	27.9	58.4	23.5	62.3		21.5		71.2	0.538	
1010	17.0	23.0	51.3	20.6	59.0		21.6		77.2	0.566	†

REMARKS.—Friction, 12.7 lb. = 18.1 F.M.E.P.; water, 72.5° C.

* 12.5 amps. † Occasional missing.

LEYLAND ENGINE.

SERIES V. TEST NO. 2.

Consumption and Power at Half Load.

Fuel—Alcohol 95 Vol. per cent. *S.G.* 0.8155.

Barometer 30.05in.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 1060 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1000	11.6	28.4	59.6	23.9	61.0		19.0	4.5	49.7	0.758	84.5*
Alcohol											
1005	11.4	28.6	59.4	24.1	61.0		19.0	,,	33.0	1.132	†
1000	11.2	28.8	59.7	24.1	59.2		18.6	,,	34.8	1.074	‡
1000	12.5	27.5	57.8	23.4	58.8		18.5	,,	37.5	1.026	§
1000	14.0	26.0	55.7	22.5	58.8		18.7	,,	39.7	1.008	‖
1010	16.0	24.0	52.9	21.4	59.5		18.7		41.2	1.021	¶

REMARKS.—Friction, 13.4 lb. = 18.9 F.M.E.P.

* 12.5 amps. † Very occasional missing. ‡ Steady, no missing. § Very occasional flutter in exhaust: ‖ Missing. ¶ Popping back.

LEYLAND ENGINE.

SERIES V. TEST NO. 3.

Consumption and Power at Half Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 5 per cent.** *S.G.* 0.813.
Barometer 30.05in.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 1060 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1000	11.5	28.5	59.8	23.9	64.0		19.0	4.5	45.4	0.829	84.5*
Alcohol											
1005	10.7	29.3	60.4	24.5	63.5		18.5	..	32.8	1.120	†
1000	10.6	29.4	60.5	24.5	62.3		18.7	..	35.0	1.050	
1010	11.3	28.7	59.5	24.3	62.8		..		36.9	1.004	‡
1040	13.1	26.9	57.0	23.2	61.8		19.0	..	39.8	0.975	§
1015	13.9	26.2	56.0	22.8	61.0		19.0	..	41.5	0.951	‖
1000	16.6	23.4	52.0	21.0	59.9		19.0		43.5	0.986	¶

REMARKS.—Friction. 13.4 lb.; water temp., 73.5° C.

*12.5 amps. † Occasional missing. ‡ Very occasional missing. § Steady. ‖ Very occasional popping back. ¶ Missing and popping back.

LEYLAND ENGINE.

SERIES V. TEST No. 4.

Consumption and Power at Half Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 10 per cent.** *S.G.* 0.810.

Barometer 30.05in.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ‡ pint		Heat input, 1060 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
1000	11.7	28.3	59.4	23.8	59.8		18.2	4.5	44.3	0.854	84.5*
Alcohol											
1000	11.2	28.8	59.6	24.1	67.5		19.6	,,	32.7	1.142	†
1000	10.9	29.1	60.1	24.3	64.2		20.0	,,	34.5	1.074	‡
1010	11.3	28.7	59.5	24.1	64.0		20.1	,,	36.9	1.012	‡
1000	11.8	28.2	58.8	23.8	64.5		,,	,,	39.8	0.950	‡
1010	13.3	26.7	56.7	22.9	63.0		20.0	,,	41.8	0.940	‡
1010	16.3	23.7	52.4	21.2	61.5		,,	,,	45.6	0.931	§

REMARKS.—Friction as per Test No. 3 = 13.4 lb.

* 12.5 amps. † Occasional missing. ‡ Steady. § Occasional fluttering in exhaust.

LEYLAND ENGINE.

SERIES V. TEST No. 5.

Consumption and Power at Half Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 15 per cent.** *S.G.* 0.8055.

Barometer 30.05in.

Revs. per min.	Torque at 3ft. rad. W = 40 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 1060 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
995	12.3	27.7	58.9	23.3	59.7		20.8	4.5	42.7	0.905	84.5*
Alcohol											
1000	11.0	29.0	59.9	24.2	64.0		21.0	..	33.3	1.117	†
1000	10.9	29.1	60.1	24.3	64.4		21.0	..	36.5	1.015	
1000	11.2	28.8	59.6	24.1	65.0		21.0	..	39.5	0.946	‡
1000	12.7	27.3	57.5	23.2	64.0		21.0	..	41.4	0.937	‡
1000	16.4	23.6	52.3	21.1	62.0		21.0	..	46.6	0.915	§
1000	19.1	20.9	48.5	19.6	59.2		21.0	..	48.9	0.939	‖

REMARKS.—Friction as per Test No. 3, 13.4 lb.

* 12.5 amps. † Steady; miss if richer. ‡ Steady. § Very occasional missing. ‖ Miss badly.

LEYLAND ENGINE.

SERIES V. TEST NO. 6.

Consumption and Power at Quarter Load.

Fuel—**Petrol "A."** *S.G.* 0.785.

Barometer 29.97in.

Revs. per min.	Torque at 3ft. rad. W = 24 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 750 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
											Volts
1000	12.0	12.0	36.7	14.7	75.0		21.0	4.5	68.2	0.897	71.5*
1000	15.0	9.0	32.4	13.0	60.0		21.0	,,	61.4	1.125	†
1000	14.2	9.8	33.5	13.5	56.0		21.0	,,	63.7	1.048	†
1005	12.9	11.1	35.4	14.3	62.0		21.0	,,	67.6	0.931	‡
1010	11.6	12.4	37.2	15.1	67.0		21.0	,,	73.6	0.810	‡
1000	10.7	13.3	38.5	15.5	69.0		21.0	,,	77.6	0.748	‡
1000	11.0	13.0	38.1	15.3	66.0		22.0	,,	82.4	0.714	
1000	11.1	12.9	37.9	15.2	66.5		21.5	,,	89.4	0.662	§
1010	11.3	12.7	37.8	15.3	67.0		21.5	,,	93.6	0.629	‖
1000	12.3	11.7	36.2	14.5					101.0	0.614	

REMARKS.—Friction, 13.8 lb., water outlet 75° C.

* 10.5 amps. † Occasional fluttering in exhaust. ‡ Very occasional fluttering.
§ Missing. ‖ Frequent missing.

LEYLAND ENGINE.

SERIES V. TEST NO. 7.

Consumption and Power at Quarter Load.

Fuel—**Alcohol 95 Vol. per cent.** *S.G.* 0.8155.

Barometer 29.92in.

Revs. per min.	Torque at 3ft. rad. W = 24 lb.		Temperature ° C.					Ignition position No.	Fuel consumption. Unit vol. ½ pint		Heat input, 750 watts
	Spring balance	Nett weight (= w)	I.M. E.P.	Ind. H.P.	Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"*A*"										Volts
995	8.9	15.1	41.1	16.4	62.5		22.0	4.5	67.2	0.912	73*
Alcohol											
1000	9.4	14.6	40.3	16.2	64.0		22.5	,,	47.7	1.166	†
1000	9.2	14.8	40.4	16.3	64.0		22.5		51.3	1.078	‡
1000	10.2	13.8	40.6	15.7	61.8		23.0		55.5	1.033	‡
1010	11.4	12.6	32.2	15.2	61.0		23.2		57.1	1.038	§
1005	12.1	11.9	36.5	14.8	61.3				60.2	1.010	§
990	16.2	7.8	30.7	12.2	62.0		23.0		65.7	1.120	‖

REMARKS.—Friction test taken at 13.8 lb.

* 10.9 amps. † Quite steady; very occasional missing. ‡ Quite steady. § Slight occasional fluttering in exhaust. ‖ Frequent missing.

LEYLAND ENGINE.

SERIES V. TEST NO. 8.

Consumption and Power at Quarter Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 5 per cent.** *S.G.* 0.813.
Barometer 29.92in.

Revs. per min.	Torque at 3ft. rad. W = 24 lb.		I.M.E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 750 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"*A*"										Volts
1000	7.7	16.3	42.7	17.7	66.0		24.0		90.4	0.563	72*
Alcohol											
1010	10.2	13.8	39.2	15.9	69.5		23.6	4.5	45.8	1.233	†
1000	9.9	14.1	39.8	15.9	65.5		23.6	,,	48.2	1.168	‡
1000	9.0	15.0	40.9	16.5	65.0		23.6	,,	50.6	1.071	‡
1005	11.0	13.0	38.1	14.2	64.0		23.8		61.3	1.031	†
1005	17.4	6.6	29.0	11.7	61.0		23.9		69.9	1.100	

REMARKS.—Friction test taken at 13.8 lb.

* 10.5 amps. † Very occasional missing. ‡ Steady.

LEYLAND ENGINE.

SERIES V. TEST No. 9.

Consumption and Power at Quarter Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 10 per cent.** *S.G.* 0.810.

Barometer 29.92in.

Revs. per min.	Torque at 3ft. rad. W = 24 lb.		I.M. E.P.	Ind. H.P.	Temperature ° C.			Ignition position No.	Fuel consumption. Unit vol. ¼ pint		Heat input, 750 watts
	Spring balance	Nett weight (= w)			Water outlet	Inlet housing	Engine room		Seconds unit vol	Pint per I.H.P. per hr.	
1	2	3	4	5	6	7	8	9	10	11	12
Petrol	"A"										Volts
995	8.7	15.3	41.3	16.5	65.0		23.8	4.5	68.3	0.794	72*
Alcohol											
1010	11.0	13.0	38.1	15.5	64.2		23.8	..	46.4	1.250	†
1000	9.4	14.6	40.3	16.2	65.5		..	..	49.2	1.131	
1005	8.2	15.8	42.0	17.0	66.0		23.8	..	52.9	1.000	
995	8.6	15.4	41.5	16.7	65.5		24.0	..	56.2	0.958	
1000	9.9	14.1	39.6	15.9	64.8		24.0	..	60.4	0.937	
1000	11.6	12.4	37.2	14.9	60.0		..	..	65.2	0.925	‡
1000	14.0	10.0	33.8	13.6	55.5		24.2	..	67.8	0.978	

REMARKS.—Friction test, 13.4 lb.; temp., 69.8° C. Friction taken as 13.8 lb. as above test doubtful.

* 10.5 amps. † Missing. ‡ Very occasional missing.

LEYLAND ENGINE.

SERIES V. TEST No. 10.

Consumption and Power at Quarter Load.

Fuel—**Alcohol 95 Vol. per cent and Ether 15 per cent.** *S.G.* 0.8055.

Barometer 29.92in.

Revs. per min.	Torque at 3ft. rad. W = 24 lb. Spring balance	Nett weight (= w)	I.M.E.P.	Ind. H.P.	Temperature ° C. Water outlet	Inlet housing	Engine room	Ignition position No.	Fuel consumption. Unit vol. ‡ pint Seconds unit vol	Pint per I.H.P. per hr.	Heat input, 750 watts
1	2	3	4	5	6	7	8	9	10	11	12
Petrol "A"											Volts
1000	8.6	15.4	40.5	16.7	65.5		22.9	4.5	80.5	0.670	72*
Alcohol											
1000	9.0	15.0	40.9	16.1	66.0		22.9	,,	48.7	1.145	†
1005	8.6	15.4	41.5	16.8	65.8		23.6	,,	52.5	1.020	‡
1000	9.1	14.9	40.7	16.4	,,		,,	,,	56.4	0.974	‡
1000	10.7	13.3	38.5	15.5	62.0		22.8	,,	60.4	0.961	‡
1000	13.3	10.7	36.2	14.0	60.5		22.6	,,	64.9	0.993	§
1000	16.4	7.6	30.4	12.2	58.6		22.6	,,	70.0	1.050	‖

REMARKS.—Friction, 14.5 lb. at 69° C. Friction taken as 13.8 lb. as above test doubtful.

* 10.6 amps. † Steady; misses if richer. ‡ Steady. § Occasional fluttering in exhaust. ‖ Missing.

SECTION IV.

THE WORK OF MIDGLEY AND BOYD ON DETONATION.

SOLUBILITY RELATIONS AND OTHER PROPERTIES OF MIXED FUELS.

FURTHER NOTES ON FUEL RESEARCH.

THE WORK OF MIDGLEY AND BOYD.*

By A. E. Dunstan, D.Sc., F.I.C., F.C.S., and Stephen Card, B.Sc.

The Bouncing-Pin Apparatus.

Midgley and Boyd have investigated, on behalf of the General Motors Research Corporation, the detonation characteristics of a fuel by means of what they call their bouncing-pin apparatus, Fig. 1, which they claim to be a method of measuring the "amount of detonation" occurring with a particular fuel. The apparatus consists of an opening in the head of the combustion chamber into which is screwed a long cylinder fitted with a piston, the underside of which is flush with the inner surface of the cylinder head, and which is held down by a powerful spring. On the upper side of the piston there rests, merely by gravity, a heavy steel pin, and immediately above the top of the pin there are a pair of contact-points connected in series with a source of current, a lamp, and an electrolytic cell containing dilute sulphuric acid. During normal combustion of the fuel the pressure of the spring is so adjusted that the piston moves up and down only a very small distance (a few thousandths of an inch). When the explosion is a detonating one, the sudden production of a very large pressure causes a greater movement of the piston, and the bouncing-pin is thrown clear, thus closing the circuit. In the course of a few minutes, if the detonation is fairly frequent, a considerable amount of gas will be collected in the cell, and the lamp may glow continuously. Midgley assumes that the volume of gas collected in a given number of revolutions, with the engine having the same adjustment, is proportional to the "amount of detonation" produced by the fuel.

* The material gathered together here has been obtained from the following sources :—

T. Midgley, jun., and T. A. Boyd, J. Ind. Eng. Chem., 1922, No. 7, p. 589.

T. Midgley, jun., and T. A. Boyd, Nat. Pet. News, July 5th, 1922 (J. Soc. Automotive Engineers, June, 1922).

T. Midgley, jun., J. Ind. Eng. Chem., 1922, No. 9, p. 849.

T. Midgley, jun., J. Ind. Eng. Chem., 1922, No. 10, p. 894.

T. Midgley, jun., J. Ind. Eng. Chem., 1923, No. 4, p. 421.

Oil and Gas Journal, April 10th, 1924, p. 126.

The Lamp (Magazine of Standard of N.J.), Feb., 1924, p. 19.

To enable measurements to be taken by the use of this apparatus it is necessary that there shall always be some detonation, and therefore Midgley has used an easily-detonating mixture of hydrocarbons, namely, kerosene, as the basic fuel. He has stated that no two explosions are precisely similar, and that although detonation is often occurring, not every explosion is a detonating one. The volume of gas collected in a given time is regarded as an integration of the "detonation" over that period.

The actual engine used was a small lighting set, the generator of which serves to give an indication of the power output. The engine

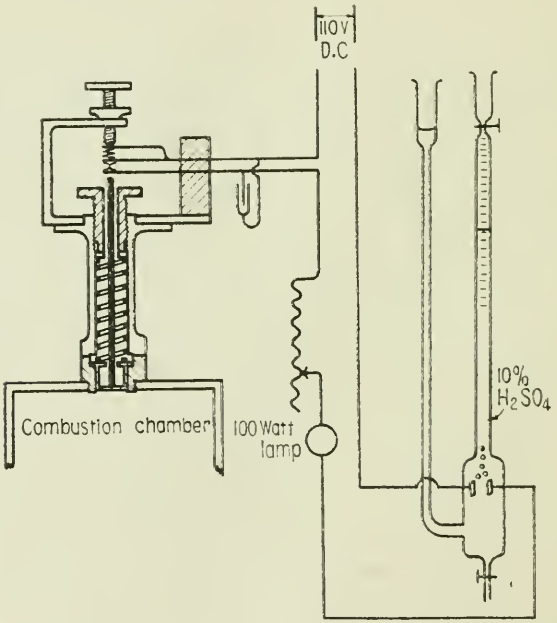


FIG. 1.—Diagrammatic representation of bouncing-pin apparatus.

has a single air cooled cylinder of 2.5in. bore by 5in. stroke, provided with four detachable heads recessed to different amounts so that compression-ratios of 3.47 : 1, 3.87 : 1, 4.59 : 1, and 5.36 : 1 can be obtained.

Method of Test.

When examining the anti-detonating effect of a fuel, Midgley has used xylidine as the standard, comparing mixtures of xylidine in

kerosene with mixtures of benzol, toluol, xylol, and alcohol in kerosene. He gives the specific case of the mixture containing 45 per cent by volume of benzene and 55 per cent of kerosene. A compression-ratio of 3.87 : 1 was used, which was high enough to allow some detonation, but not enough to cut down the power output seriously or cause the engine to run erratically. In all cases the mixture was adjusted to give maximum detonation, which mixture is stated to be approximately the weakest which will give maximum power. The fuel system of the engine was arranged in duplicate so that rapid changes from one fuel to another could be made while the engine

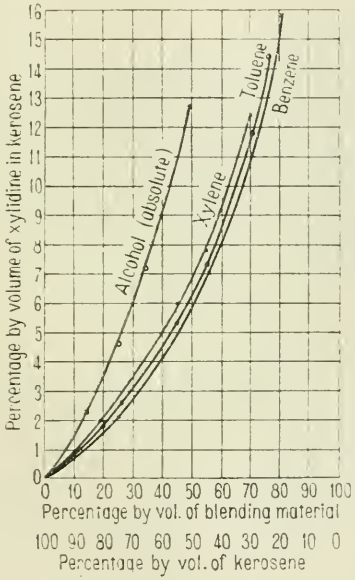

Percentage by volume of xylidine in kerosene
Alcohol (absolute)
Xylene
Toluene
Benzene
0 10 20 30 40 50 60 70 80 90 100
Percentage by vol. of blending material
100 90 80 70 60 50 40 30 20 10 0
Percentage by vol. of kerosene

FIG. 2.—Comparison of alcohol, benzene, toluene and xylene.

was running. By trial it was found that a 5 per cent xylidine/kerosene fuel had a slightly less detonating tendency than the benzene/kerosene fuel under examination. Keeping conditions in the engine the same, alternate one-minute runs were made on each fuel and the volume of gas collected was measured in each case. The benzene/kerosene blend was then replaced by a 4 per cent xylidine/kerosene mixture and the process was repeated. By plotting the average amounts of gas produced using the three fuels against the percentages of xylidine in the standard mixtures the exact xylidine mixture corresponding to the given benzene mixture could be determined.

Results.

The relative values of the aromatic hydrocarbons and absolute alcohol as suppressors of detonation were then determined, the results being given in Fig. 2.

From the results it appears that toluene is slightly better than benzene and xylene slightly better than toluene as an anti-detonating substance. It may be noted here that Ricardo found toluene the most effective of the aromatic hydrocarbons, and that it was much better than benzene as a suppressor of detonation. Absolute alcohol appears much better than the aromatic hydrocarbons, on the volume-for-volume basis.

Anti-knock Substances.

Using the same apparatus, Midgley found that there was a variety of substances which could suppress detonation, sometimes even when present in very small amounts, and these he called anti-knock substances. On the other hand, he discovered several substances which could induce detonation. Most of the substances he mentions are volatile liquids and they can either be dissolved in the fuel or the vapour can be allowed to be sucked in with the air.

Table I. shows the effects of some of the more noteworthy substances :—

TABLE I.

Element.	Form in which used.	Effect on detonating quality of fuel.
Iodine	Element	Suppresses detonation
Bromine	Element	Induces detonation
Oxygen	Element	Induces detonation
Oxygen	Ethyl nitrite or nitrate	Induces detonation
Nitrogen	Aniline	Suppresses detonation
Selenium, Tellurium	Di-ethyl compounds	Suppresses detonation
Tin, Lead	Tetra-ethyl compounds	Suppresses detonation

Using benzene in kerosene as the standard fuel mixture, Midgley examined the relative efficiencies of the anti-knock substances. The comparison is given in Table II. The proportions, both by volume and by molecules, of these substances which produced an anti-detonating effect equivalent to 25 per cent by vol. of benzene mixed with the kerosene are given.

The very small amount of some of these substances which will cause considerable anti-detonating effect is remarkable, and has been demonstrated by some striking experiments. If an un-stoppered

bottle containing lead tetra-ethyl is held near the air intake of a violently detonating engine, then the small amount of the vapour sucked in with the air causes the detonation to cease and the power output rises. If the engine now be run on a mixture containing 35 per cent benzene there will be little or no detonation, but if an unstoppered bottle containing some volatile nitrate or nitrite (such as isopropyl nitrite) be held close to the air-intake, then violent detonation will occur and it may be of sufficient severity to stop the engine.

The marked effect of lead tetra-ethyl on the combustion of an easily-detonating mixture such as that of acetylene and air has also

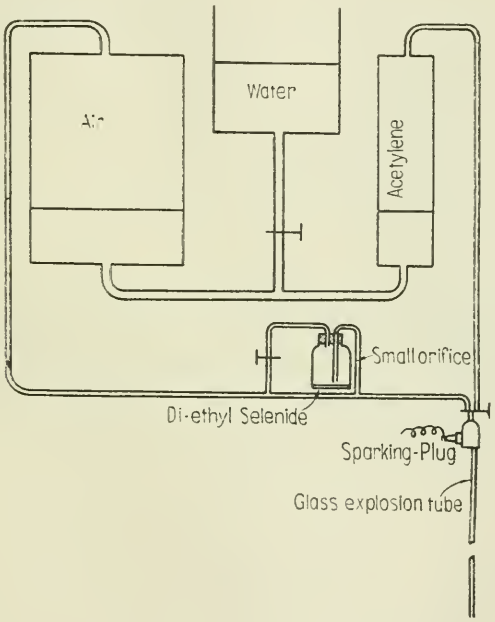


FIG. 3.—Apparatus to show effect of anti-knock agent on explosion of acetylene and air.

been demonstrated. The apparatus consists of two vessels, Fig. 3, one containing acetylene and the other air. The cross-sections of these vessels are such that when the gases are displaced by water from a single tank they are admitted to an open-ended explosion tube in such proportions that the resulting mixture is the one producing maximum detonation. There is provision for sparking the mixture at the closed end of the tube, and, if the gases are now ignited, a detonating explosion will develop, accompanied, as is usual, by a

sharp " crack " and a high degree of luminosity of the flame. Some inches may even be blown off the end of the glass explosion tube.

If the experiment is now repeated with the exception that the entering air is allowed to carry with it a little of the vapour from a bottle containing di-ethyl selenide, in the manner shown in the diagram, then the resulting explosion is quiet and non-detonating,

TABLE II.

Element.	Compound.	Corresponding per cent. by volume.	Corresponding number of molecules of theoretical air/fuel mixture to one molecule of anti-knock.
	Benzene	25	150
Iodine	Ethyl iodide	1.6	2,150
Nitrogen	Xylidine	2.0	2,600
Tin	Tin tetra-ethyl	1.2	7,100
Selenium	Di-ethyl selenide	0.4	11,750
Tellurium	Di-ethyl telluride	0.1	50,000
Lead	Lead tetra-ethyl	0.04	215,000

although the amount of di-ethyl selenide present constitutes only one seven-hundredth part of the volume of the gaseous mixture.

Midgley regards these anti-knock materials as quite distinct from fuels such as the aromatic hydrocarbons, though, broadly speaking, their effect is the same. The aromatic hydrocarbons are themselves fuels which possess anti-detonating properties, and when mixed with

TABLE III.

Element.	Reciprocal of number of gram.-mols. required to give anti-knock effect equivalent to 1 gram.-mol. of aniline.	
	Ethyl compound.	Phenyl compound.
Iodine	1.09	0.88
Selenium	6.9	5.2
Tellurium	26.8	22.0
Oxygen	−0.036 (induces knocking)	0.122

kerosene they impart these properties to the fuel in proportion to the amount of aromatic added. The anti-knock substances, however, are efficient when present in extremely minute proportions. Table II. showing that lead tetra-ethyl is quite efficient when present to the extent of one molecule in 200,000 molecules of the air/fuel mixture, in this case the molecules of lead tetra-ethyl being 1400 times as

efficient as those of benzene. Also the effect does not increase in proportion to the amount of anti-knock added, but reaches a maximum with relatively high concentrations.

In these anti-knock substances the detonation-suppressing characteristics are ascribed to the particular atom present, *e.g.*, to the lead atom in lead tetra-ethyl and to the nitrogen atom in the aromatic amines. Table III. gives a comparison made by Midgley of the effects of various atoms with two different groups attached, which seems to show that it is a function of the atom.

Action of Anti-knock Materials.

Since such small amounts of anti-knock produce such a marked effect on combustion, Midgley considers that the action bears some resemblance to that of a catalyst—a kind of negative catalyst, reducing the velocity of the chemical action of combustion. He considers that is their sole effect, and has suggested that they may act as screens selectively absorbing radiation and thus reducing the flame-velocity, reasoning on the lines of Perrin's radiation theory. As regards the relationship between flame-velocity and detonation, Midgley bases his reasoning on the fact that the pressure of the gas immediately in front of the flame (P_1) is greater than the pressure behind it (P_2). These pressures are dependent on the velocity of combustion and in normal combustion the value of the pressure-difference is small, of the order of 1 lb. per sq. in. By mathematical reasoning, Midgley infers that there is a certain critical value of P_2, and therefore a certain critical value of the flame-velocity, at and above which the pressure-difference suddenly becomes very great. He considers an explosion under these conditions to be a detonating one, the sharp "pinking" sound being due to the dense mass of gas immediately in front of the flame and at high pressures, striking the top or sides of the combustion chamber.

Application.

There were several difficulties to contend with before the use of lead tetra-ethyl could become a practicable proposition.

In the first place, it was a rare chemical costing over £100 per pound, and it was only after a great deal of experimenting and the introduction of large-scale production that the cost was able to be reduced so that the price is about 10s. per pound. As the quantity of the anti-knock added is 3 c.c. per American gallon of gasoline, this difficulty has been removed. Then the preliminary trials showed that the products of combustion had a very harmful effect on the sparking-plug electrodes, causing them to disappear. This difficulty also has been obviated by the addition of some halogen-containing compound.

Apparently carbon tetra-chloride has been used, but at present it is stated that ethylene di-bromide is employed. The function of

these compounds is to convert the lead into lead halides immediately after combustion, and so prevent its action on the plug electrodes. Furthermore, lead tetra-ethyl itself (or its break-down products) is stated to affect the lubricating oil very seriously. As a consequence engines running on " ethyl gas " are found to collect a greyish deposit on the walls of the combustion chamber, though it is stated that no harm ensues.. Again, lead tetra-ethyl, which is a colourless liquid of 1.62 S.G. with a boiling point of 152° C. and miscible with gasoline in all proportions, decomposes on exposure to light and air, forming at first a white precipitate which later turns brown. Like other compounds of lead, it is poisonous.

The anti-knock is sold through the medium of specially fitted petrol pumps. These pumps are normal except that a small apparatus, called the " ethylizer," is attached. If ethylized gasoline is required, the apparatus can be put in connection with the pump by operating a small lever, and every gallon of gasoline delivered will automatically have added to it the requisite quantity of lead tetra-ethyl and ethylene di-bromide. Fresh supplies of the " ethyl " are supplied in sealed metal bottles containing about a quart, which are inverted and dropped in the top of the ethylizer, a pointed tube puncturing the seal and allowing the anti-knock to run out as required.

SOLUBILITY RELATIONS AND OTHER PROPERTIES OF MIXED FUELS.

W. R. Ormandy and E. C. Craven.

It has been shown in the earlier parts of this volume that great differences exist between various constituents of motor fuels in connection with their power of withstanding compression in an engine without detonating. Alcohol is about twice as efficient as commercial motor benzole per weight unit in reducing the tendency to detonation, but, unfortunately, only absolute alcohol is miscible with paraffins, naphthenes, and aromatics in every proportion. The strongest alcohol of commerce other than the small amount specially prepared for chemical industries contains some water, and indeed the strongest alcohol which can be prepared by the ordinary systems of distillation has only about 95 Vol. per cent of the strength of absolute alcohol, the remainder being water. This arises from the fact that a mixture of alcohol in water in the proportion of 95.57 : 4.43 by weight distils at a lower temperature than any other mixture of these two ingredients, and is called an azeo-tropic mixture. As questions of alcohol strength are expressed sometimes in the form of degrees over proof and at other times as a weight relationship, and again as a volume relationship, the figures in Table I., taken from Thorpe's Tables, are given for future reference.

Even though water-free alcohol could be obtained, the fact that such alcohol is hygroscopic renders it desirable that the solubility relationship of alcohol containing various quantities of water to the possible constituents of motor fuels should be referred to. The situation is rendered the more complex by the fact that the solubility of aqueous alcohol in all the ordinary constituents of motor fuels is greatly affected by temperature, and therefore the temperature relationships with regard to the solubility of alcohol, water, and petrol constituents must be reviewed. As all petrols and even motor benzoles are complex mixtures, it will be well to study first the behaviour of some comparatively pure hydro-carbons in regard to ethyl alcohol and water. As the whole subject has been dealt with at length elsewhere,* it is only necessary here to indicate general results.

* See J.I. Pet. Tech., Ormandy and Craven, Vol. 7, p. 423, 1921; Vol. 8, p. 181, 1922.

TABLE I.

Specific gravity in air at 60°F./60° F. or 15.6° C./15.6° C.	Percentage of ethyl alcohol.		Percentage of fiscal proof spirit.
	By weight.	By volume at 60° F. or 15.6° C.	
0.79359	100.00	100.00	175.35
0.7940	99.87	99.92	175.21
4	99.75	99.84	175.08
8	99.62	99.76	174.94
0.7950	99.55	99.72	174.87
4	99.43	99.64	174.73
8	99.29	99.56	174.59
0.7960	99.22	99.52	174.52
4	99.10	99.44	174.38
8	98.97	99.36	174.23
0.7970	98.90	99.32	174.16
4	98.78	99.24	174.02
8	98.64	99.16	173.88
0.7980	98.57	99.12	173.80
4	98.45	99.04	173.66
8	98.31	98.95	173.52
0.7990	98.24	98.91	173.44
4	98.12	98.83	173.30
8	97.98	98.75	173.15
0.8000	97.91	98.70	173.07
4	97.79	98.62	172.93
8	97.67	98.54	172.78
0.8010	97.59	98.49	172.71
4	97.46	98.41	172.56
8	97.32	98.33	172.41
0.8020	97.25	98.28	172.33
4	97.12	98.20	172.19
8	96.98	98.11	172.03
0.8030	96.91	98.06	171.95
4	96.78	97.98	171.80
8	96.64	97.89	171.64
0.8040	96.57	97.84	171.56
4	96.44	97.76	171.41
8	96.30	97.66	171.25
0.8050	96.23	97.62	171.17
4	96.10	97.54	171.01
8	95.96	97.44	170.85

TABLE I. (*continued*).

Specific gravity in air at 60° F./60° F. or 15.6° C./15.6° C.	Percentage of ethyl alcohol.		Percentage of fiscal proof spirit.
	By weight.	By volume at 60° F. or 15.6° C.	
0.8060	95.89	97.39	170.77
4	95.76	97.31	170.61
8	95.62	97.21	170.45
0.8070	95.55	97.16	170.37
4	95.41	97.08	170.21
8	95.27	96.98	170.04
0.8080	95.20	96.93	169.96
4	95.06	96.84	169.80
8	94.92	96.74	169.64
0.8090	94.85	96.69	169.55
4	94.71	96.60	169.39
8	94.57	96.50	169.22
0.8100	94.50	96.45	169.13
4	94.36	96.36	168.97
8	94.22	96.26	168.80
0.8110	94.15	96.21	168.71
4	94.01	96.12	168.55
8	93.87	96.02	168.37
0.8120	93.80	95.97	168.28
4	93.66	95.87	168.12
8	93.51	95.77	167.95
0.8130	93.44	95.72	167.86
4	93.31	95.62	167.68
8	93.17	95.52	167.50
0.8140	93.08	95.47	167.41
4	92.94	95.37	167.23
8	92.80	95.27	167.05
0.8150	92.72	95.22	166.96
4	92.58	95.12	166.78
8	92.44	95.02	166.60
0.8160	92.36	94.97	166.51
4	92.22	94.87	166.33
8	92.08	94.77	166.15
0.8170	92.00	94.71	166.06
4	91.86	94.61	165.88
8	91.71	94.51	165.69

Fig. 1 shows the solubility of pure benzene in alcohol with varying percentages of water at various temperatures, and Fig. 2 gives corresponding results for commercial xylols. Experiments on the aromatic bodies show that the more volatile of the series are more readily soluble, but even in the case of the most volatile of them —benzene—it is necessary to have alcohol of strength 88.4 per cent by weight in order to render stable a mixture of 80 parts alcohol and 20 parts benzene, while for a 90 : 10 alcohol, benzene mixture the strength must be 92.3 weight per cent and these at 15° Centi-

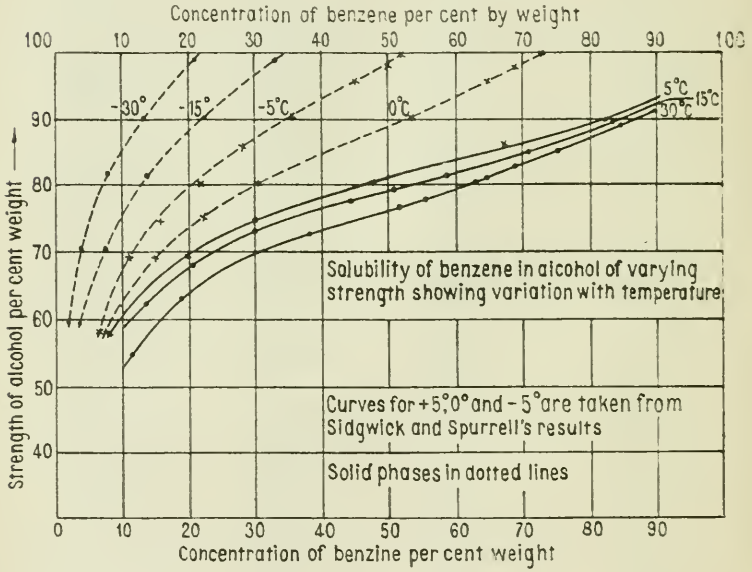


Fig. 1.

grade. The curves show how great is the influence of temperature. The corresponding figures for toluene at 15° C. are :—

80 alcohol 20 toluene requires alcohol 90.5 per cent weight strength.
90 alcohol 10 toluene requires alcohol 93.5 per cent weight strength.

It is much easier to get pure aromatics than pure paraffins, but, nevertheless, the results obtained with the many samples of the latter are sufficient to enable some general conclusions to be drawn. The curves in Fig. 3 relate to solubilities in alcohol of 92 per cent by weight, this being the strong alcohol of commerce, and it will be seen that one pentane sample was very soluble ; this contained

a lot of dissolved gases and bodies of very low molecular weight, and bears out the general relation that the more volatile and lower molecular weight paraffins are more soluble in alcohol with water content than the less volatile and heavier members of the group. The temperature effect is greater the greater the solubility of the paraffin.

As heptane may be taken as the typical petrol paraffin, the solubility relations of this body at two temperatures are given in Fig. 4.* The discovery that alcohol raises the detonation pressure for equal

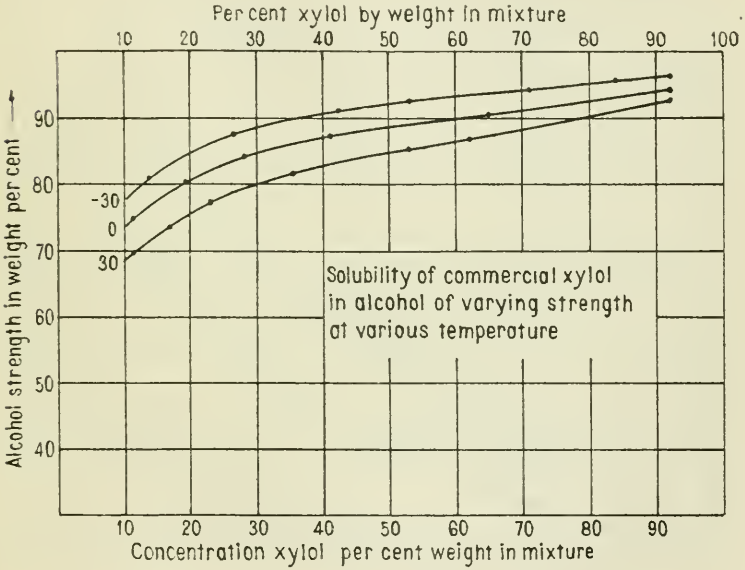


Fig. 2.

weights about twice as much as commercial benzole brings up the question as to the conditions under which small quantities of alcohol can be dissolved in petrols.

The curves in Fig. 5 relate to the petrols used in the foregoing research and are consequently of greater practical interest than the academic figures on pure compounds. Fuel A is rich in aromatics and naphthenes and owes its high solubility thereto in spite of the comparatively high boiling point of the spirit. Fuel B, though low in aromatics, is very light—99 per cent boiling below 120° C. Fuels

* See Ormandy and Craven, *loc. cit.*

E and F are typical paraffin petrols of medium boiling range, and H, although chiefly naphthenic, has such a high boiling range that the

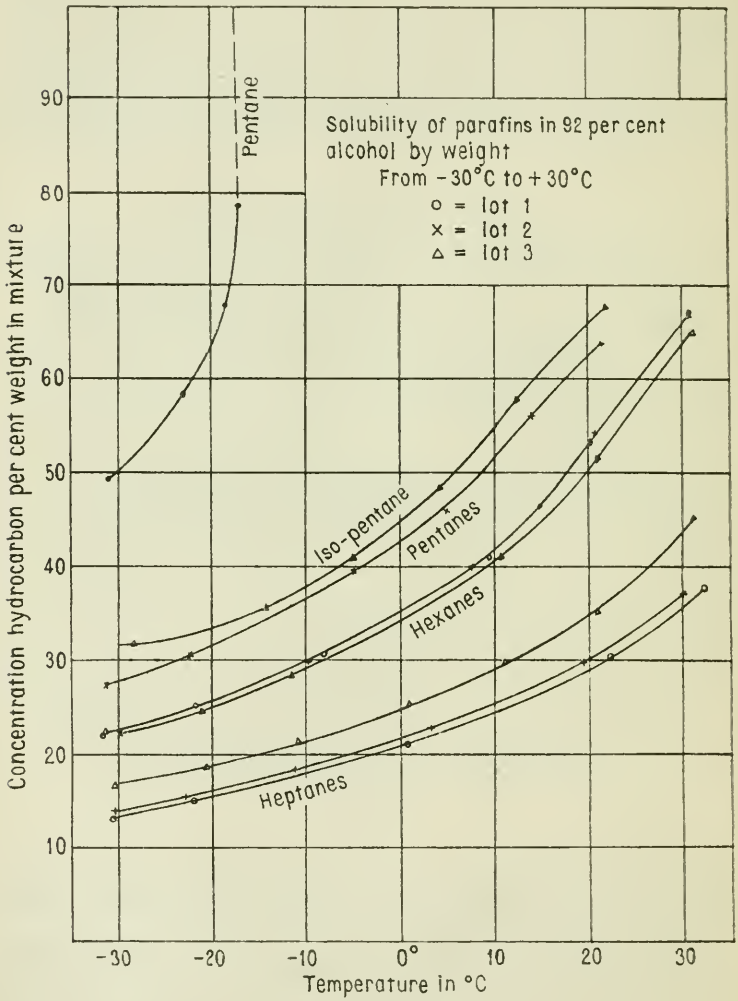


Fig. 3.

advantage of the greater solubility of the naphthenes over the paraffins of the same boiling range is lost to sight.

Table II. gives a good oversight of the influence of constitution, volatility and temperature on the behaviour of various fuels. but it must be read in conjunction with the particulars of fuels given on page 142 (*ante*) *et seq.*

Mixing Agents.

The comparatively low solubility of commercial alcohol in petrols raises the question as to whether any bodies exist which, when added in traces. will bring about increased solubility or stability. Many experiments have been made and some of the results are

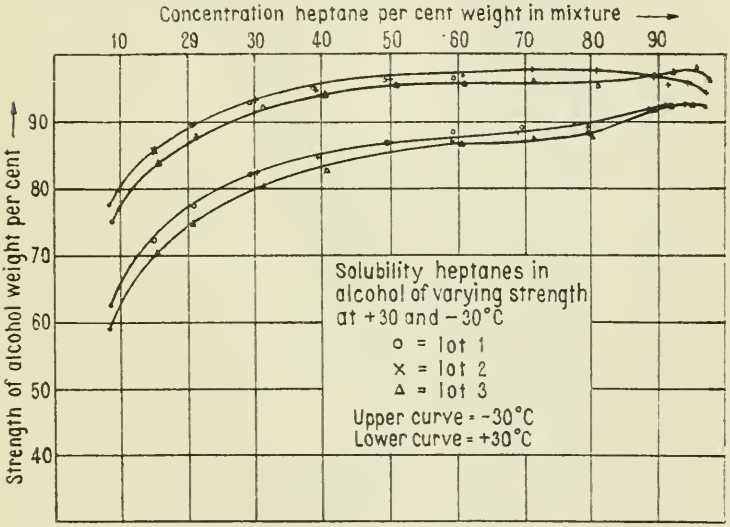


FIG. 4.

embodied in Table III., which applies in most cases to mixtures containing 85 Vols. hydro-carbon, 10 Vols. 95 Vol. alcohol, 5 Vols. mixing agent, and expresses the results in relation to heptane and a typical No. 1 petrol.*

No body yet examined is one-fifth part as effective in stopping separation as water is in bringing it about, nor does it seem likely that any such body will be found. It is interesting to note that the best mixing agents consist of hydro-carbon groups combined with an hydroxyl group. Amyl alcohol is the best mixing agent yet investigated, but Fig. 6 shows how little influence even this exerts. To add 10 per cent of alcohol to a Pratts No. 1 spirit before and after

* See J.I. Pet. Tech., Ormandy and Craven, Vol. 9, No. 36.

adding 5 per cent amyl alcohol, the spirit would have to be 95.8 and 93.2 weight per cent strong at 15° Centigrade. The addition of mixing agent 50 per cent in amount of the ethyl alcohol employed has only reduced the necessary strength 2.6 weight per cent.*

Since mixing agents may be ruled out except in so far as these are fuels and obtainable in quantity, the problem of procuring strong—

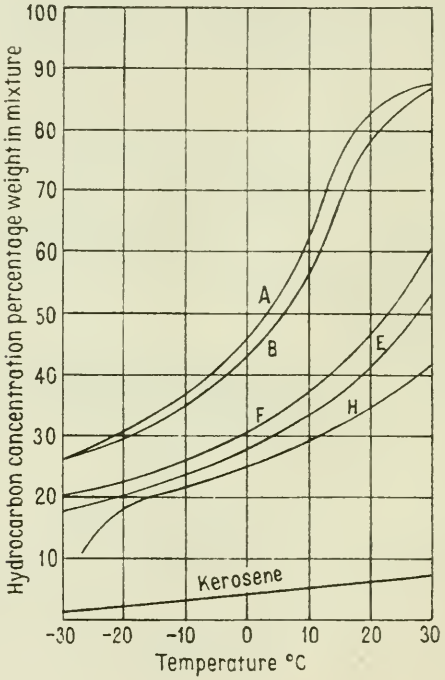


FIG. 5.—Solubility of Ricardo standard petrols in 92½ per cent by weight of alcohol at various temperatures.

indeed almost absolute—alcohol becomes of importance. Among the many methods employed or suggested are :—

1. Distilling off quicklime.
2. Distilling the vapours through quicklime.
3. Removing the water by distillation with a third body forming an azeotropic mixture.
4. By differential diffusion.

* See Proc. I.A.E., Vol. XVI., Part II., p. 143, Ormandy and Craven.

5. By distilling the vapours through bodies like potassium acetate in alcoholic solution and then drying down the potassium acetate for re-use.
6. Freezing out the water from the alcohol after addition of petrol, benzole, or the like.
7. Distilling off glycerine.

No. 1 is a laboratory method unsuited for large-scale use.

No. 2 is used on a large scale and is practical and economical where good burnt lime is available.

TABLE II.—**Collected Figures for Solubility of Alcohol in Petrols. The Figures give the Strength of Alcohol Necessary to Dissolve to the Extent Stated.**

Wt. Concentration A.	No. 1, Pratt's.	A.	B.	E.	F.	H.	Hexane.	Heptane.
2	96.1	95.4	96.2	96.3	96.1	96.4	95.0	96.0
4	96.2	95.2	96.2	96.2	96.0	95.8	95.2	96.2
30° C. 6	95.8	94.7	96.0	95.7	95.8	95.4	95.4	96.4
8	95.5	94.3	95.0	95.4	95.4	95.2	95.3	96.3
10	95.2	94.1	94.4	95.2	95.2	95.1	95.0	96.1
12	95.0	93.9	94.1	95.0	95.0	94.9	94.6	95.8
2	—	98.1	97.6	98.3	97.7	97.6	98.5	98.0
4	—	98.1	98.2	98.7	98.0	98.9	98.1	98.5
6	—	97.9	98.3	98.7	98.2	99.0	97.8	98.8
30° C. 8	—	97.6	97.8	98.5	98.2	98.8	97.5	98.9
10	—	97.3	97.2	98.1	98.1	98.7	97.1	98.9
12	—	97.0	96.6	97.7	98.0	98.7	96.8	98.7

No. 3 is the method long employed in Germany and originated by Professor Young and described in his well-known work on distillation.

No. 4 seems to be in the experimental stage, but would be of value in places where chemical operations would be difficult.

No. 5 is used on a commercial scale, but little seems to be known as to the details.

No. 6 necessitates the cooling of large bulks of liquid to remove comparatively little water.

No. 7 is said to be employed in France on a commercial scale.

Literature regarding many of the above processes can be found in the volume of proceedings of the Congress on Combustible Liquids, held by the Société de Chimie Industrielle in Paris in 1923.

Toluene Number and Solubility.

An examination of the facts so far brought to light shows that, of the groups paraffins, naphthenes and aromatics, the ability to

withstand compression is greatest at equal boiling point for the aromatics and least for the paraffins, and further, that in two of these groups the ability to withstand compression goes down as the molecular weight goes up. The same relationship holds for the alcohol solubility figures, but a fuller examination of these relations on a wider basis goes to prove that, however true the similarity may be for the lower numbers of the three groups so far examined, it is not true for a wide range of substances of different character. Thus ether has a high solubility, but withstands a very low compression-

Table III.

K = solubility factor.

Mixing agent.	Heptane.	Petrol No. 1.
Benzene	3	1
Toluene	—6	—
m-Xylene	—18	—
Turpentine	0 about	< 0
Heptane	—11	—
Cresol	123	114
Anilin	51	26
Methyl alcohol	—24	—30
Ethyl ,, 95 vol.	0	0
Ethyl ,, 100 ,,	72	72
Propyl ,,	90	70
Butyl ,,	168	144
Amyl ,,	170	150
Hexyl ,,	47	—
Ether	Small	3
Acetone	41	18
Ethyl acetate	Small	27
Cyclo-hexanol	201	150
Acetaldehyde	—20	—
Water	—1000	—

pressure. Carbon disulphide is very soluble, but its resistance to detonation is only medium and aniline with high solubility is very resistant to compression.

Toluene Number and Spontaneous-ignition Temperature.

The work of Moore, Constam and Schlaepfer, and Krupp's research department has shown that there is some regularity in the behaviour of groups of chemical bodies when tested in the spontaneous-ignition meter. Here again we find the same general order. The paraffins ignite readily—that is, at low temperatures—and the heavier the paraffin the lower the ignition temperature. This applies also to the naphthenic and aromatic groups, and the general order of the groups

is that of resistance to compression —*i.e.*, toluene number. Once again, however, a closer examination shows that the relationship is strictly limited, not exactly true even in the groups and not by any means quantitative. On testing bodies outside the three named groups the discrepancies become more pronounced, nor can a relationship between solubility and spontaneous-ignition temperature be proved, although there would appear to be some within the three groups. This is shown by Fig. 7, which is self-explanatory.* Generally speaking, there is some relation between ignition tempera-

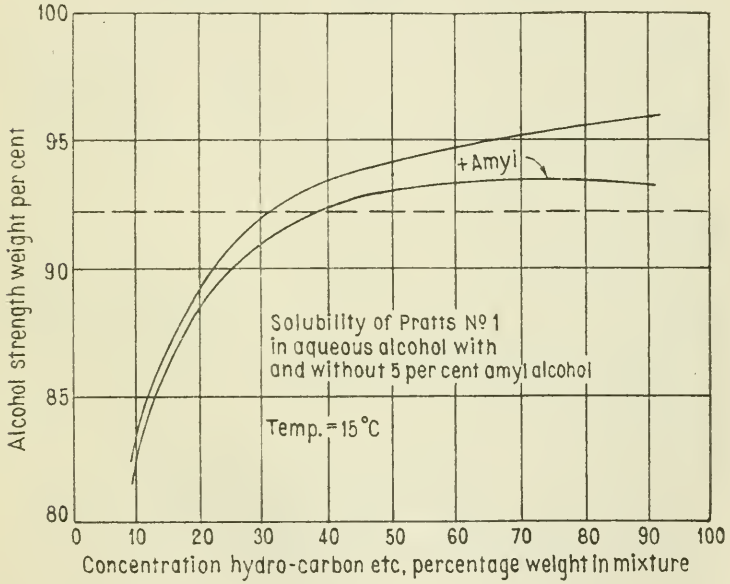


FIG. 6.

ture and resistance to compression within the three groups, but the quantitative nature of the relationship is all wrong. The discovery by Ricardo of substances which are much influenced by compression and not much by temperature-differences and others which are influenced in the opposite sense is of great interest and should lead to some fundamental explanation at present lacking. The discovery by Midgley and Boyd of bodies which, when present in traces, retard detonation is also of great interest. It is unfortunate that these are all poisonous and for the most part very unpleasant-smelling bodies.

* See J.I. Pet. Tech., Vol. 7, No. 28, 1921, Ormandy and Craven.

Lead tetra-ethyl used in conjunction with carbon tetra-chloride or ethylene bromide would lead to the production of about 5 grammes of lead chloride or bromide per gallon of fuel burnt, and as lead salts are cumulative poisons the results of large-scale employment might lead to trouble. Since the general order of spontaneous-ignition temperatures among the three groups is that of their resistance to compression, it was of interest to try the effect of the addition of small amounts of lead tetra-ethyl to heptane. In view of the effect of one part in 2000 on detonation, it might have been expected to

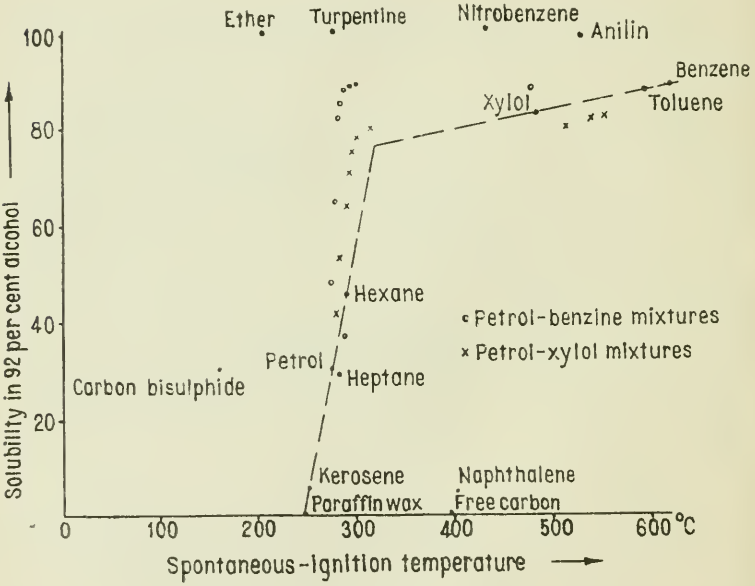


FIG. 7.

raise the spontaneous-ignition temperature, but, on the contrary, this was lowered by 14 degrees C., and amyl nitrite and nitrate, which Midgley and Boyd found to bring about early detonation, proved to raise the ignition temperature by about 4 degrees.*

The actual mechanism of the action of bodies such as lead tetra-ethyl and amyl nitrite is still to seek. To speak of "negative catalysis" is merely to hide the mystery in other words.

Some years ago the authors suggested that selective picking up of radiant energy might have something to do with the

* Fuller details of this work by Ormandy and Craven will appear shortly in the J.I. Pet. Tech.

remarkably different behaviour of paraffins and aromatics in the spontaneous-ignition meter and indeed in the engine cylinder. The absorption spectra in the infra-red of these bodies gives some ground for this suggestion. Unfortunately, the necessary apparatus for this kind of work is very expensive and progress is consequently slow. Under full-throttle conditions, owing to the high temperature of the exhaust valve and piston head, the conditions in the cylinder bear some resemblance to the crucible in the spontaneous-ignition meter, and so, too, in the fact that the fuel in both cases is supplied in the liquid state. For this reason an examination of the conditions in the ignition meter near the critical temperature seemed called for. The not-yet-published work, previously referred to, deals with this and leads to the conclusion that the "wait period," which may amount to half a minute, is one of flameless combustion, during which the temperature is gradually rising, leading to eventual explosion if the temperature is over the critical temperature, but quite measurable for many degrees below the critical temperature even when no explosion occurs.

The obvious suggestion of surface combustion is not borne out by experiments made in the presence of bodies such as platinum, silver, copper, quartz, or porcelain, either in the massive form or in some cases in powder. It is hard to believe that if a surface effect were in question there should be no appreciable difference under such varied conditions, yet such is the case. It was thought of interest by Craven and Pond to try the effect of passing heptane or toluene and oxygen in about the theoretical proportions for complete combustion over iron in an electrically-heated silica tube in order to find out the temperature at which carbon dioxide would be produced. This was found to occur at 240° C., which is not far removed from the spontaneous-ignition temperature of 241° C.

In the case of toluene having a spontaneous-ignition temperature of 596° C., however, the first evidences of CO_2 were found at 210° C. Obviously the matter wants a lot of further investigation. There is little doubt that flameless combustion must take place in the cylinder, in many cases even during the compression stroke.

Equilibrium Boiling Point.

The temperature shown in the foregoing Ricardo experiments by a thermometer placed in the induction pipe is considered to give indications of some value and to be related in some way to the temperature at which 50 per cent of the fuel would have distilled over under standard Engler conditions. It may be suggested that, since the object of good carburation is to get the whole of the fuel in the vapour form, at any rate by the end of the compression stroke, a figure of interest would be the temperature at which the sum of the partial pressures of all the numerous constituents of the average fuel

would together amount to 760 mm. pressure, supposing that this could be estimated without fractionating the fuel at the same time. This problem was studied by Keith and Whatmough in designing their ingenious carburettor, and further work on the equilibrium boiling point and latent heat of mixed fuels, particularly the special fuels made up of petrols, benzole, acetone, ethyl and methyl alcohol and water used in racing, might be referred to.*

Conclusions.

At a very early stage of the considerations of the Empire Motor Fuel Committee it became evident that in addition to the work to be done by Ricardo, many problems of a chemical and physical nature required investigation, and, thanks to the financial assistance of the Distillers Co., Ltd., and the Department of Scientific and Industrial Research, the authors of this chapter were able to investigate a number of problems, and the preceding notes are an attempt to abstract such portions of this work as have the most direct bearing upon the investigations comprised in the whole volume.

A list is appended of these publications, some of which may prove of interest to workers in special fields.

"The Relation between Alcohol-Water Solubility and Spontaneous-Ignition Temperatures," J.I. Pet. Tech., 1921, 7, 325.

"The Systems Ethyl Alcohol-Water-Aromatic Hydro-carbons from 30° C. to — 30° C.," J.I. Pet. Tech., 1921, 7, 422.

"The Systems Ethyl Alcohol-Water-Paraffins from plus 30° C. to — 30° C.," J.I. Pet. Tech., 1922, 8, 181.

"Note on Solubility of Benzene in Weak Alcohol," J.I. Pet. Tech., 1922, 8, 213.

"Note on the Effect of Small Quantities of Volatile Impurities on the Flash-Point of Kerosene" (Craven and Banks), J.I. Pet. Tech., 1922, 8, 490.

"Further Investigations in the Physico-Chemical Significance of Flash-Point Temperatures," J.I. Pet. Tech., 1923, 9, 33.

"A Method for Expressing the Value of Mixing Agents between Hydro-carbons and 95 Vol. per cent Alcohol," J.I. Pet. Tech., 1923, 9, 129.

"Equilibrium Boiling Point and Latent Heat of Vaporisation of Motor Fuels," J.I. Pet. Tech., 1923, 9, 368.

"Some Further Studies with the Moore Spontaneous-Ignition Meter," J.I. Pet. Tech. In press.

"The Effect of Thermometric Lag in Engler Distillation," J.I. Pet. Tech. In press.

"The Physical Properties of Motor Fuels," Proc. I.A.E., Vol. XVI., Part II., p. 143.

* See J.I. Pet. Tech., Ormandy and Craven, Vol. 9, No. 39.

FURTHER NOTE ON FUEL RESEARCH.

By H. R. Ricardo.

Since the completion of the researches described in the preceding pages further work has been done which is germane to the subject, of which the results are herein briefly set forth, so that the extent and the limitations of our knowledge at the present day may be appreciated.

As regards detonation, undoubtedly the most important recent work is that of Midgley on the effect of the organo-metal compounds, of which lead tetra-ethyl, the basis of the "ethyl fluid" marketed in the U.S.A., is the best known—see p. 305 *ante*. The author has made some experiments on these compounds on the variable-compression and on other research engines. On the variable-compression engine the following results were obtained :—

Fuel.	Highest useful compression-ratio.
Aromatic free petrol, pure	4.85
,, ,, + 0.1 per cent lead tetra-ethyl	6.05
,, ,, + 0.2 per cent lead tetra-ethyl	6.85
Petrol "I." pure	4.3
,, ,, + 0.1 per cent lead tetra-ethyl	5.5

Percentages by volume.

These figures give a "toluene value" for this substance of about 40,000, as compared with 67 for benzene. The ratio between the "anti-knock" properties of these two substances is thus $\frac{40{,}000}{67}$ = 597, which agrees well with Midgley's figures of $\frac{25}{0.04}$ or 625, considering the difference in procedure and apparatus. No explanation or beginnings of an explanation appear yet to have been found as to the precise behaviour of these organic compounds, which, unlike the aromatic compounds, appear to have no influence whatever either upon the rate of flame propagation or upon the "self-ignition temperature" of the fuel.

The contamination of the lubricant with lead salts appears, how-

ever, to be at present a serious trouble. Thus in some experiments on a 4in. by 6in. water cooled single-cylinder unit comparative endurance tests were made on petrol (Shell 1) with and without "ethyl fluid" in the proportion of 7½ c.c. per gallon, or 1 in 605 by volume. This quantity was sufficient to raise the detonation point, in the variable-compression engine, by 1.2. The duration of each test was 27 hours, the speed being 900 revs. per minute and the brake horse-power 5.5. At the end of each test the engine was dismantled and the carbon deposit collected and weighed. With "ethyl fluid" the deposit on the piston top was of a slate colour and weighed 2.45 grams, of which 27.6 per cent was incombustible ash. With undoped petrol the deposit was 1.8 grams of the usual black colour, of which only 1.9 per cent was ash. The lubricant around the piston rings was, in the former test, clearly contaminated with lead salts, having the appearance of grey paint, but the guide surfaces were fairly clean and the percentage of ash in the oil was not increased.

A similar test was also made with lead tetra-ethyl alone. In this case the deposit was so severe that the test was discontinued after 13½ hours, the piston being thickly coated with a white deposit, which had so contaminated the oil on the rubbing surfaces that these latter were practically dry.

It is clear from this that the halogen compounds in "ethyl fluid" considerably reduce the deposit, but until it is completely eliminated really satisfactory running is improbable, as contamination of the oil by solid particles cannot be tolerated.

*Further experiments have also been made as to the effect of adding inert diluent gases to the charge. It has long been known that such gases tend to suppress detonation, and cooled exhaust gases have in practice been used to dilute fuels such as kerosene and coke-oven gas, which detonate at low compression-ratios. In these experiments the three constituents of exhaust gas, nitrogen, carbon dioxide, and steam were taken separately, and their effect on detonation and range of burning observed. Oxygen was also tested as a "negative diluent." The engine employed was the variable-compression unit used in the fuel research, the diluents being admitted through holes in the carburettor choke-tube, or to the air intake. The carbon dioxide and nitrogen were supplied from a calibrated gas-holder, while the steam was generated in a small electrically-heated boiler, calibrated by measurement of the steam supply with various electrical inputs. This method was found to give accurate results and to be very convenient in operation. Apart from these modifications the procedure was the same as in the fuel tests recorded earlier, a check test with no diluent being made simultaneously with each test on a diluted charge.

* See *The Automobile Engineer*, June, 1924.

From the results of these tests the explosion pressures and temperatures at the ratio at which detonation commenced in each case were calculated employing the methods and data used by Tizard and Pye.* They are based on somewhat arbitrary assumptions as to heat-losses, etc., and thus may not be of a very high absolute accuracy, but relatively to one another they should not be far wrong.

The results obtained by the above methods are given in Table I.

TABLE I.—*Calculated Explosion Temperatures and Pressures with Various Diluents.*

1	2	3	4	5	6
Diluent.	Percentage by Vol.	Highest useful compression-ratio.	Flame temperature, ° C.	Explosion-pressure, lb. per sq. in. gauge.	Rise of pressure on ignition, lb. per sq. in.
Nil	—	4.5	2450	450	360
CO_2	14.1	5.55	2150	491	365
	19.6	6.15	2050	520	366
N_2	17.1	5.0	2260	461	357
	27.15	5.50	2130	481	358
H_2O	2.8	4.65	2415	458	365
	4.5	4.85	2390	473	374
	8.9	5.10	2325	485	378
O_2 (= N_2	3.08 — 13.05)	4.12	2590	435	356

NOTES.

Column 2.—This figure = $\frac{\text{Vol. diluent at N.T.P.}}{\text{Vol. (fresh mixture + residual exhaust) at N.T.P.}}$

Column 4.—Assumed heat-loss to jacket on ignition 6 per cent.

Column 5.—Pressure at end of suction stroke assumed 14 lb. per sq. in. absolute.

Combustion is, in all cases, assumed to be complete at top dead-centre.

From these figures it will be noted that when diluents are added :—

(1) The flame temperature of detonation falls considerably.
(2) The explosion pressure, on the other hand, rises rapidly.
(3) The rise of pressure during ignition, *i.e.*, explosion-pressure minus compression-pressure is practically constant, within the errors of experiment, when either nitrogen, oxygen, or CO_2 is the diluent. With steam this figure rises.

Referring to Figs. 1 and 2, where the flame pressures and temperatures respectively at the detonation point are shown plotted against

* *Ante.*

compression-ratio, it is seen that the nitrogen, CO_2, and oxygen figures lie practically on the same curve, whereas the steam figures are uniformly higher. In other words, steam is considerably more effective in checking detonation, relatively to nitrogen or CO_2, than can be accounted for by the difference in its thermal properties. From this it appears probable that some chemical, or at any rate non-thermal, characteristic is operating in this case. The thermal equivalence of nitrogen and CO_2 renders it probable that with these gases no non-thermal characteristic, or only a small one, is operative,

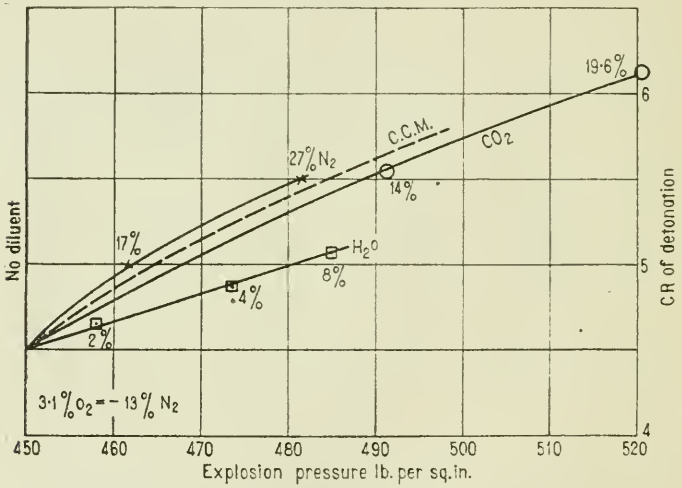


FIG. 1.—Theoretical explosion pressures at observed detonation points with various diluents.

C.C.M. Calculated curve for exhaust products of complete combustion mixture.

since it is unlikely that any chemical property would be the same in two gases differing so widely in nature.

The Causes of Detonation.

Several theories have, from time to time, been put forward as to the nature of the factors which are responsible for detonation, and it is of interest to examine them in the light of the figures given herein. The influence of the nature of the fuel need not, of course, be taken into consideration in this case, since in these tests one fuel only was used. The earlier theories, that detonation is a function of the temperature and pressure of compression, are clearly untenable, since neither of these factors can be affected appreciably by a slight

dilution of the charge with a cold inert gas. The theory which has received most support in recent times is that put forward by Tizard, which states that detonation is a function of the flame temperature, *i.e.*, that for any given fuel there is a definite flame temperature at which detonation commences. With increase in compression-ratio the dilution of the fresh charge by residual exhaust gases decreases, owing to the reduced clearance-space, and the flame temperature consequently rises. The effect of change of compression-ratio upon detonation is thus explained, as is also the comparatively

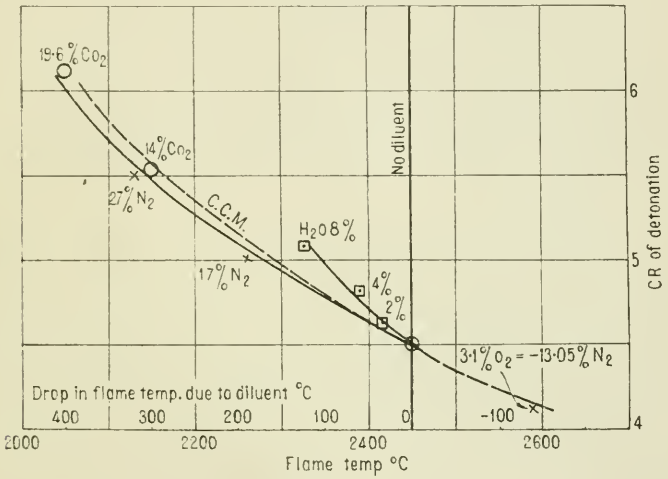


FIG. 2.—Theoretical flame temperatures at observed detonation points with various diluents.

C.C.M. Calculated curve for exhaust products of complete combustion mixture.

slight effect of change in the induction temperature. It also accounts for the elimination of detonation by throttling, since in this case the proportion of residuals in the charge increases and the flame temperature consequently falls.

This latter case is of particular interest, since the flame temperatures with various degrees of throttling can be estimated, and Tizard has been able to find, in some experiments which he has previously published,* some quantitative confirmation of his theory.

The temperature, and therefore the density, of the residuals is, however, a little uncertain, and this renders such calculations somewhat speculative.

* Proc., N.E. Coast Inst. of E. and S. Vol. XXXVII.

What this theory does not explain is the fact that when the atmospheric pressure is diminished, as in the case of an aero-engine at high altitudes, the tendency to detonate also decreases. In this case the flame temperature is unaltered, except to a slight extent by change in the jacket loss and surrounding atmospheric temperature, since the pressure of both fresh charge and residuals is reduced equally, so that detonation should not be appreciably affected, whereas the effect is, in actual fact, very considerable.

If we examine this theory in the light of the quantitative data now available, Table I. makes it clear that flame temperature is not the sole determining factor, since it varies considerably, for the same degree of detonation, under different conditions. It might, perhaps, be argued that the diluent was exerting a chemical effect analogous to that of the fuel, but it is, as before stated, improbable that two gases of such differing nature as nitrogen and CO_2 should have a non-thermal effect so exactly in the ratio of their thermal properties. It would appear, then, that the flame temperature is not the only factor to be considered.

The explosion-pressures likewise show no sign of uniformity, but the "explosion-pressure rise" shows a most marked constancy. Except in the case of steam where some non-thermal factor appears, as stated before, to be operative, this pressure-rise, at the detonation point, is constant to $\pm$ 4.5 lb. per sq. in., or little more than 1 per cent on an average of 360 lb. per sq. in. The corresponding flame temperatures vary from 2050° C. to 2590° C., and the explosion-pressures from 435 lb. per sq. in. to 520 lb. per sq. in., so that the remarkable constancy of the pressure-rise over six tests under such widely different conditions can hardly fail to have some significance. Whether the pressure-rise at the detonation point remains constant when the conditions are altered by other means, such as throttling or change of external air pressure, cannot at present be determined, but such a conjecture would in both cases account qualitatively for the effect observed.

It is well known, however, that heating of the charge, whether by increase in jacket temperature or by induction-pipe heating, tends to increase detonation to a small but definite extent, whereas the "explosion-pressure rise" must necessarily decrease owing to the reduced charge mass. We cannot, therefore, maintain that detonation is a function of this pressure-rise alone, and can only suppose that the observed constancy of this factor at the detonation point was due to its being, under the conditions of these tests, a function of the real determinant, which fact may afford some clue as to the latter.

Detonation, then, appears to be a function both of the temperature and pressure of combustion, but the nature of such function is not clear. Logarithmic and semi-logarithmic plotting of the "explosion" pressures and temperatures do not reveal any linear relationship,

and the relation between these two factors appears, therefore, not to be a simple one.

A point of interest about the dilution figures is that the maximum dilutions tested with each diluent represent, very roughly, the maximum amount which could be added without causing misfiring. That these proportions should be widely different on a volume basis was to be expected, but the flame temperature in each case would be expected to be approximately the same, if the effect were purely thermal. This, however, is not the case, as will be seen from Table II.

Table II.

Diluent.	Percentage by Vol.	Limiting flame temperature.	Drop in flame temperature due to dilution at limiting flame temperature.
CO_2	19.6	2050	400
N_2	27.15	2130	320
H_2O	8.9	2325	125

Since the figures only represent the limiting conditions very roughly, too much should not be made of the difference between CO_2 and N_2, but the restricted range of dilution with steam is very noticeable. Here again some non-thermal influence appears to be at work.

Evidence has also been obtained that, with very weak mixtures, such as can be utilised with hydrogen as fuel, or with a stratified charge, the tendency to detonate is much reduced, presumably on account of the low "explosion" pressures and temperatures under those conditions. This is illustrated by the results shown in Fig. 3, which is based on weak-mixture tests on hydrogen and on petrol with a stratified charge.

Range of Burning. Weak Mixture Operation.

This factor has, until recently, received comparatively little attention, as far as high-speed engines are concerned, not through a lack of appreciation of its significance, but because all known liquid fuels vary so little in this respect that the question was of small practical importance as a criterion of fuel value.

The advantages of a wide range of available mixture strength on the weak side are :—

(1) Very appreciable variations in the air/fuel ratio are, in most cases, met with as a result of carburation and distribution

errors. Thus, although the weak limit with liquid fuels, as determined on a single-cylinder engine, is found with a mixture about 20 per cent weak, *i.e.*, with sufficient fuel to combine with 80 per cent of the air, the necessity of obtaining such a mixture in every cylinder under the worst combination of carburation and distribution errors renders it necessary, with a multi-cylinder engine, to use an average mixture strength containing a definite excess of fuel, which,

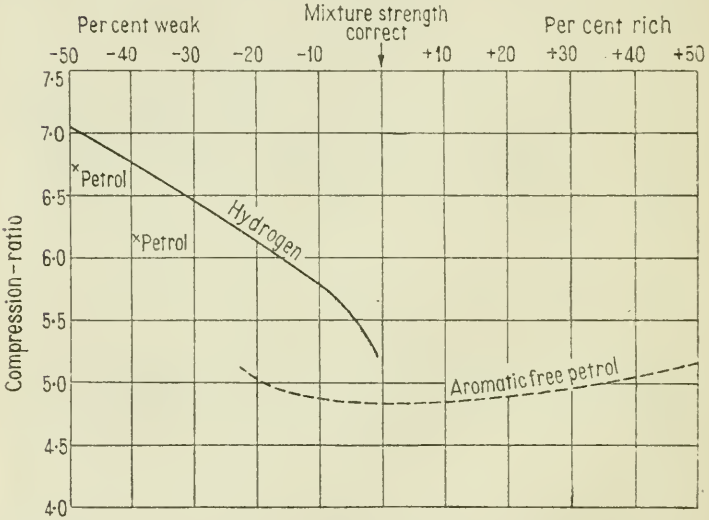


FIG. 3.—Compression-ratio ratio at which the detonation point becomes apparent with varying mixture strengths. Two points are also shown obtained with aromatic free petrol when working with a stratified charge.

of course, is very wasteful. If a weaker cylinder charge can be used this loss can be eliminated.

(2) For part-load operation a higher thermal efficiency could be obtained by "quality control," that is, by running with a full air charge and a reduced fuel supply, than by "quantity control," *i.e.*, reducing both fuel and air supply proportionately. This is on account of the lower flame temperature attained in the former case, which reduces the mean specific heat, dissociation, and direct heat-loss of the working fluid, and thus gives a higher cycle efficiency.

(3) The low flame temperature of a weak charge is of great value from a practical point of view, as it considerably reduces

carbonisation and the manifold other troubles which arise from excessive heat-flow. This is naturally of particular importance in air cooled cylinders.

With liquid fuels the available mixture range, as explained before, is far too limited to allow of quality control, the only known fuel in which this range is measurably extended being ethyl ether, of which a 22 per cent weak mixture, that is, one with fuel only sufficient to com-

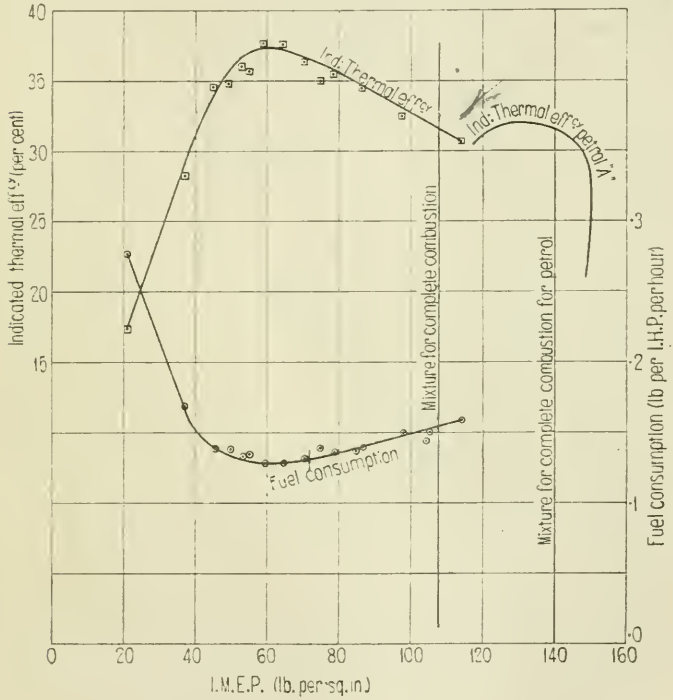


FIG. 4.—Thermal efficiency obtainable with Hydrogen.
Compression-ratio 5.45 : 1 ; 1500 revs. per minute.
Full throttle ; no heat to air intake.
Fuel, Hydrogen, supply varied.

bine with 78 per cent of the air, can be used, though this extension is too small to be of any practical value. With several gaseous fuels, more particularly coke-oven gas, the range is considerably greater, and "quality control" is to some extent practised, though the range is not sufficient to allow of an engine being controlled entirely by these means. Recent experiments, however, with hydrogen show that

with this fuel the available range is very great indeed, so much so in fact that the variable-compression research engine could be run with the throttle wide open and the power output varied from nothing to full load merely by control of the hydrogen supply.

Under these conditions some remarkable thermal efficiencies were attained. This is well brought out in Fig. 4, which shows the results obtained at a compression-ratio of 5.45 : 1, a comparative test on

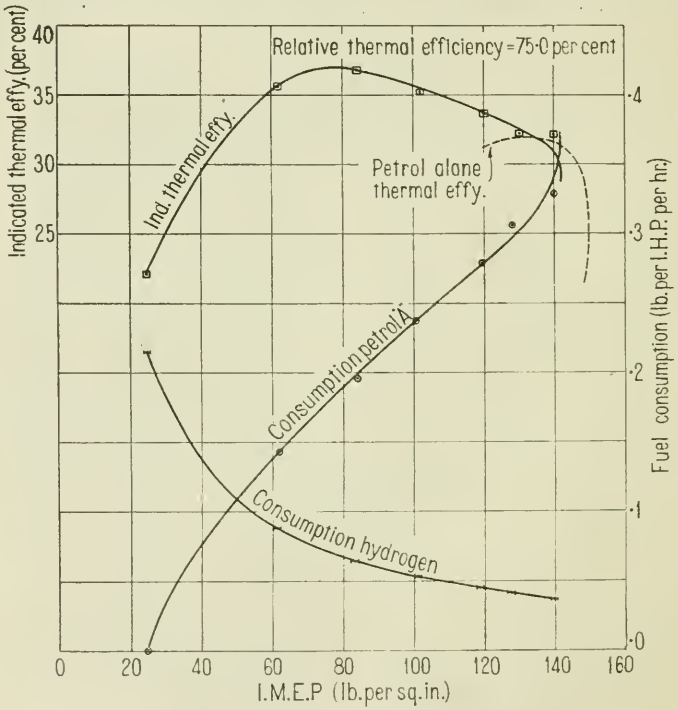


FIG. 5.—Effect of addition of Hydrogen to Petrol.

Compression-ratio 5.45 : 1; 1500 revs. per minute.
Full throttle; no heat to air intake.
Fuel, Hydrogen (supply constant) and Petrol (supply varied).

petrol being also shown. The thermal efficiency is at a maximum at 60 lb. per sq. in. indicated mean effective pressure, with about 100 per cent excess air in the charge, being 37.5 per cent, whereas the best attained with petrol was 32 per cent, with about 15 per cent excess air. At lower powers the efficiency rapidly decreases, due presumably to incomplete or retarded burning, though even with the engine idling no mis-firing or popping-back into the carburettor occurred, and the running was perfectly regular. If, however, an

attempt was made to run with a rich hydrogen-air mixture, violent pre-ignition occurred, accompanied by firing back into the carburettor, which rendered further running impossible. Even with the compression-ratio lowered to 3.8 : 1 the same thing occurred. With a 7 : 1 compression-ratio and weak mixture running was quite smooth at light loads, and the excellent indicated efficiency of 43 per cent was attained at 74 lb. per sq. in. indicated mean effective pressure, but if this power was exceeded pre-ignition commenced.

Another valuable characteristic of hydrogen is that a small quantity will act as a "priming" charge to very weak petrol-air and

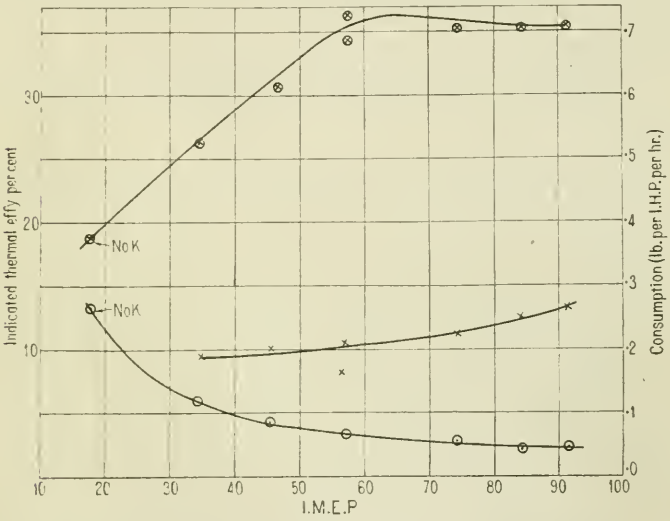


FIG. 6.—Effect of addition of Hydrogen to Kerosene.

Full throttle; 1500 revs. per minute.

Fuel, Kerosene (supply varied), Hydrogen (supply constant). Kerosene ×. Hydrogen ○.

kerosene-air mixtures. Thus in the test illustrated in Fig. 5, which was carried out in the variable-compression engine, a fixed quantity of hydrogen was admitted, just sufficient to run the engine on no load with wide-open throttle, and the power controlled entirely by means of the petrol supply. The high efficiencies attainable with hydrogen alone are, it will be seen, again realised while the maximum power output on rich mixture is attainable without any pre-ignition or popping-back. Fig. 6 shows a similar test carried out with kerosene as fuel. In this case an indicated mean effective pressure of 90 lb. per sq. in. could be carried at a compression-ratio of 5.5 : 1 with an indicated efficiency of 35.5 per cent, though severe detona-

tion occurred when running on a normal kerosene-air mixture even at a ratio of 3.8 : 1, the thermal efficiency then being only 24.5 per cent. When running at this lower ratio it was noted that the substitution of hydrogen for a part of the kerosene in the complete-combustion mixture considerably reduced the detonation, showing that hydrogen is not an easily-detonating fuel. With such non-volatile fuels the hydrogen appears, moreover, to ensure complete combustion of the kerosene, and thus eliminates the appreciable loss by incomplete combustion, which is usually a trouble of the kerosene engine. Thus at a normal mixture strength and at a compression-ratio of 3.8 : 1 the indicated thermal efficiency with kerosene alone was 24.5 per cent, and with kerosene plus about 10 per cent hydrogen 28 per cent.

This peculiarity of hydrogen, which distinguishes it so markedly from other fuels, is due presumably to its much higher flame velocity, but it is not clear why all other fuels should vary so little in this respect.

Acetylene, which has a very high rate of flame propagation, might also be expected to behave in a similar manner, but tests with this gas dissolved in acetone do not show any appreciable increase in range.

Some experiments have been made as to the effect on the available mixture range of the inert diluents of which the effect on detonation had been investigated previously. Steam, carbon dioxide, nitrogen, and oxygen were all tested, in most cases in various proportions; the result found was that in every case the minimum mean effective pressure attainable was the same. It thus appears that what limits the range is not so much the ratio of fuel to oxygen, but rather the proportion of fuel in the total cylinder charge. It is difficult, however, to determine precisely the limit of weakness with the short total range available on ordinary fuels, and it would be of considerable interest if similar experiments were carried out using hydrogen as a fuel, since the range in this case is so large that small differences would be measurable.

Stratified Charge.

Another method of extending the mixture range, well known in theory but little applied in practice, is the use of a stratified charge. This is done by admitting the charge in two separate portions, one a normal or easily-combustible mixture and the other pure air.* These are maintained in two separate layers until the mixture is ignited, when the resulting turbulence mixes the two layers, forming a weak low-temperature expanding charge. Obviously this method will, in theory, give a complete quality control, any required amount of combustible charge being admitted and the rest of the cylinder

* See p. 80 *ante*.

charged with air. The difficulty lies, however, in keeping the two charges from mixing. If they are admitted consecutively through one valve, or even to the same portion of the cylinder, almost complete mixing occurs, but it has been found that by admitting the main charge through the usual inlet valve or port and the air charge through a ring of ports at the end of the suction stroke at the bottom of the cylinder a considerable degree of stratification is attained, provided only that the excess air pressure is not more than about double that of the main charge. Fig. 7 shows a test made on a single-cylinder unit, the two curves showing the mixture range with and without stratified charge operation. In both cases the throttle was in the same position, giving, with normal operation, about half full torque. It will be seen that the weak end of the range is practically identical

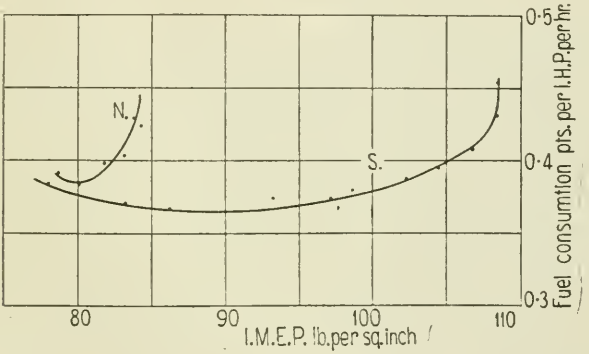


FIG. 7.—Stratification limits. Curve N, normal running; Curve S, stratified.

Fuel, Petrol A 50 per cent, Benzole 50 per cent. 1300 revs. per minute.

in each case, showing that the air charge does not appreciably dilute the mixture round the plug points, while the great extension of the rich end of the range is due to the utilisation of the supplementary air by the excess fuel in the main charge. As will be seen, there is an appreciable direct reduction in fuel consumption, but of equal significance is the enormously increased mixture range over which a high efficiency is attained. This means that in practice considerable carburation or distribution errors can exist without appreciably affecting the fuel consumption.

Ease of Starting.

It has been found that the vapour pressure of a fuel is by no means a satisfactory guide as to its starting properties. This appears to be due to the fact that in an engine the vapour is drawn from a limited

quantity of liquid, of which the final composition is therefore not that of the original fuel. Better results were obtained by finding the weakest air/fuel mixture which could be ignited by a spark after rapid compression from some fixed temperature, the compression-ratio being about 4.0 : 1. This method, though rather elaborate and troublesome, certainly appeared to give results which were directly applicable in practice.

Volumetric Efficiency.

Earlier in this volume (p. 109) it was stated that no adequate reason could be assigned for the observed decrease in volumetric efficiency with increase in compression-ratio, and at the time of writing this was correct. Recently, however, an explanation has been put forward by J. F. Alcock, of Ricardo and Co., which appears to explain adequately this peculiarity.

At the end of the exhaust stroke the clearance-space is filled with hot residual gases, and, owing to the slow motion of the piston near the top dead-centre, these hot gases are for an appreciable period not cooled to any serious extent by admixture with the fresh charge. During this period they are, of course, rapidly losing heat to the cylinder walls, and any contraction due to this which occurs after the closing of the exhaust valve will be made up by the drawing in of a corresponding volume of fresh charge, with a consequent increase in the volumetric efficiency. Now, with a low compression-ratio these residual gases are hotter, and form a greater proportion of the total charge, than with a higher ratio. Their contraction will therefore be greater with the low ratio, and the volumetric efficiency correspondingly higher, as has been seen to be the case. It should, therefore, be of advantage to take any steps possible to increase the amount of heat transfer taking place during this period. This could be done by increasing the period between the closing of the exhaust valve and the opening of the inlet valve, but mechanical considerations render any later opening of the inlet valve undesirable. Increased turbulence should improve matters by increasing the rate of heat transfer, but so many factors are influenced by a change of turbulence that it has not been found possible to separate out this effect.

The Present State of Fuel Knowledge.

Of the main fuel characteristics, detonation is undoubtedly the most important, but it is also now the one on which most data are available, though we are still in ignorance as to the fundamental factors upon which it depends. From an immediately practical point of view it may be said that the behaviour in this respect of all fuels at present available is known and that the performance of all mixtures can be predicted with reasonable certainty. Anti-knock "dopes," however, still afford a wide, and probably a profit-

able, field for investigation. Those at present known have, it is true, some serious drawbacks, but there is no reason to suppose, in the present state of knowledge, that other more practicable substances may not exist.

Fairly complete data also exist as to the influence on detonation of physical conditions such as compression temperature, flame temperature, etc., form of combustion chamber, position of sparking-plug, turbulence, etc., but at present we have no quantitative knowledge as to the effect of charge density on detonation. A determination of this effect would be of considerable value as rounding off our knowledge of the physical conditions which influence detonation. When this has been done further progress must be looked for from the purely theoretical and chemical side.

Mixture Range.

Hydrogen, the one known fuel on which a really adequate range is available, is, unfortunately, impracticable for most uses. Except in the unlikely event of some liquid fuel of similar properties being discovered, or some manner of transporting hydrogen in a compact and economical form being found, stratification seems the only hope in this direction. It seems to be impracticable to rely altogether on "quality control." It is possible, however, to do so down to about half-load, below which point the charge mass must be reduced. This range, however, is quite sufficient to effect a very substantial increase in efficiency, while eliminating the effects of carburation and distribution errors.

Distribution.

This is the subject above all others on which information is at present most required. Even with the limited mixture range ordinarily available, it is possible to attain in practice quite good efficiencies in a single-cylinder engine, but with a multi-cylinder engine it is almost invariably necessary to enrich the mixture considerably in order to make sure that the cylinder reached by the weakest mixture is getting, under all conditions, enough fuel to run on. Unfortunately, no multi-cylindered engines suitable for research have yet been constructed, and thus there are practically no data on this most vital question. Distribution, in fact, is probably the subject on which research is at present most urgently needed, and at the same time it bids fair to prove one of the most difficult problems to tackle in a comprehensive manner. In this connection, however, the use of a stratified charge and some form of mean-pressure indicator would appear to have very great possibilities as a means of determining directly and quantitatively the air/fuel ratio in each individual cylinder under actual running conditions, and seems likely to prove by far the most hopeful method of attacking the problem.

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NOTE.—Page numbers marked with a * refer to diagrams and not to the text.
Those marked *a* refer to tests on Petrol.
.... *b* .... Benzene.
.... *c* .... 90 Vol. per cent Alcohol.
.... *d* .... 95 Vol. per cent Alcohol.
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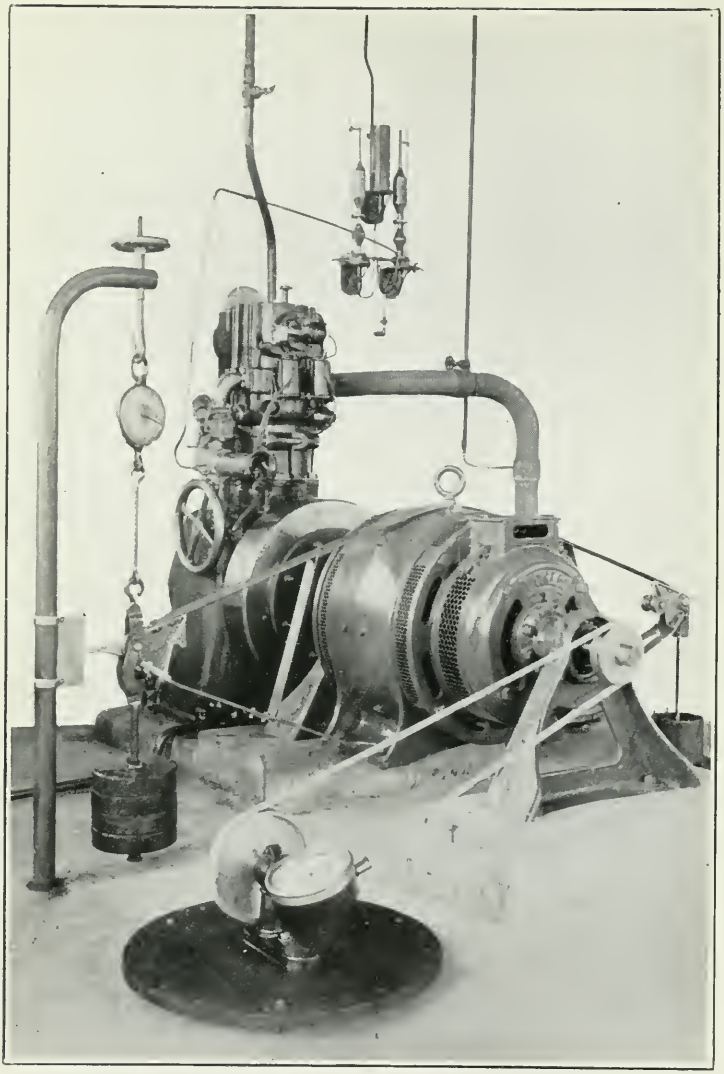

Fig. 4.—Variable-compression engine.

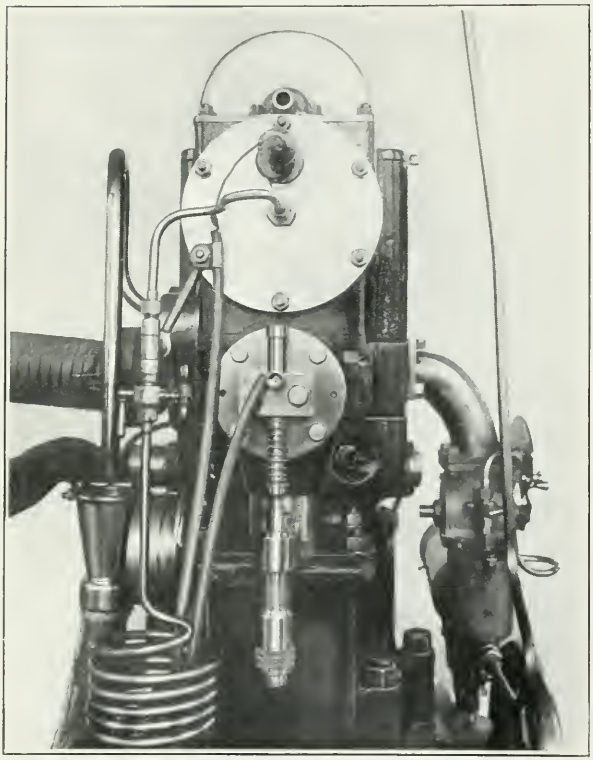

FIG. 5.—View of cylinder head of variable-compression engine. The micrometer shown is for adjustment of the cylinder to various compression-ratios.

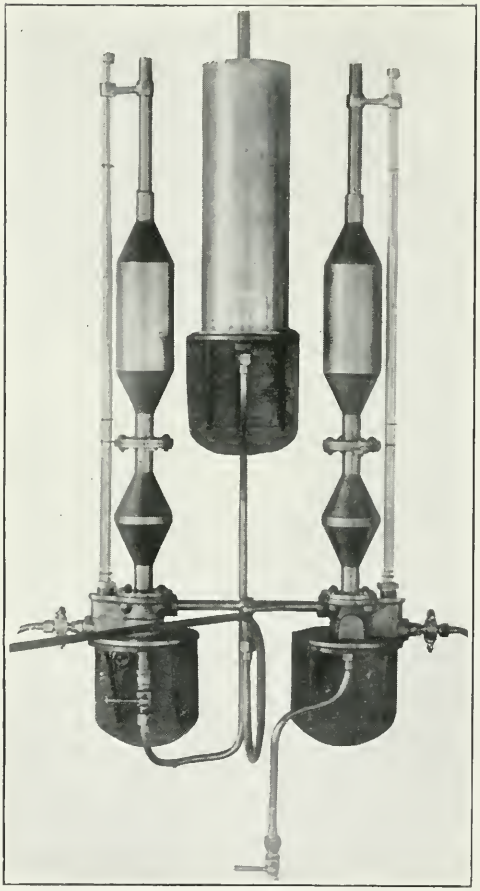

FIG. 6.—Fuel-measuring device.

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